INVESTIGATION OF THE A-7 HOOK POINT ATTACHMENT BOLT LOOSENING PROBLEM

Allen William Roessig

NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

Investigation of the A-7 Hook Point
Attachment Bolt Loosening Problem

by

Allen William Roessig, Jr.

December 1974

Thesis Advisor M. H. Bank

Approved for public release; distribution unlimited.

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Investigation of the A-7 Hook Point Attachment Bolt Loosening Problem		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis; December 1974
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Allen William Roessig, Jr.		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		12. REPORT DATE December 1974
		13. NUMBER OF PAGES
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office) Naval Postgraduate School Monterey, California 93940		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

A-7 Hook point attachment bolt
Self loosening of threaded fasteners
Loosening due to plastic strain

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Loosening of the hook point attachment bolt in the aircraft arresting hook assembly of the A-7 attack aircraft has caused a potentially dangerous situation and has demanded considerable maintenance effort. This report reviews the history of the loosening problem and attempted solutions to it. Research into a hitherto neglected possible causal mechanism, self loosening, is discussed. Experiments designed to evaluate the contribution of

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE
(Page 1) S/N 0102-014-6601

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

this new factor to the problem are reported. The conclusions drawn from these tests are: that loosening occurs as a result of plastic strain of the bolt, that the crushing of the washer also causes loss of torque, and that the existence or non-existence of a self loosening contribtion could not be established.

Investigation of the A-7 Hook Point

Attachment Bolt Loosening Problem

by

Allen William Roessig, Jr.
Lieutenant, United States Navy
B.S., Marquette University, 1967

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN AERONAUTICAL ENGINEERING

from the
NAVAL POSTGRADUATE SCHOOL
December 1974

ABSTRACT

Loosening of the hook point attachment bolt in the aircraft arresting hook assembly of the A-7 attack aircraft has caused a potentially dangerous situation and has demanded considerable maintenance effort. This report reviews the history of the loosening problem and attempted solutions to it. Research into a hitherto neglected possible causal mechanism, self loosening, is discussed. Experiments designed to evaluate the contribution of this new factor to the problem are reported. The conclusions drawn from these tests are: that loosening occurs as a result of plastic strain of the bolt, that the crushing of the washer also causes loss of torque, and that the existence or non-existence of a self loosening contribution could not be established.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOTATION

English

A	area
E	modulus of elasticity
e	total strain
F	force
K	equivalent elastic spring constant
L	bolt load
l_b	bolt grip length
P	side force
r	thread pitch diameter
r_1	equivalent base diameter
T	torque
w	weight

Greek

α	thread helix angle
β	thread form angle
ε	incremental strain
μ	coefficient of friction
σ	normal stress

Subscripts

B	base
b	bolt
c	clamped parts
F	friction
T	threads
w	weight

ACRONYMS AND ABBREVIATIONS

ATKRON/VA	Attack Squadron
COMNAVAIRPAC	Commander Naval Air Pacific Fleet
CVA	Aircraft Carrier
Msg.	Message
NAS	Naval Air Station
NAVAIR	Abbreviation used for NAVAIRSYSCOM in the identification number of publications they have authored
NAVAIRSYSCOM HQ	Naval Air Systems Command Headquarters
NAVAIRSYSCOMPREP PAC/LANT	Naval Air Systems Command Representative Pacific/Atlantic
UR	Unsatisfactory Report

Naval Message Date-Time Group Reference Format Example: ATKRON ONE TWO FIVE 031445Z Feb 74

ATKRON ONE TWO FIVE	The initial item in the group is the sending activity; other examples are NAVAIRSYSCOM HQ, COMNAVAIRPAC Etc.
03	First two numbers indicate the day of the month the message was sent; in this case the third day
1445Z	The next four numbers indicate on the 24 hour clock the time the message was sent. The letter following the time group indicates the time zone used. In this case it was sent at 1445 hours Greenwich Mean Time.
Feb	Month sent is always abbreviated with three letters.
74	Year sent uses only last two digits.

ACKNOWLEDGEMENT

The author wishes to express his special thanks to M. H. Bank for his patience, his counsel as thesis advisor, and his willingness to work on weekends. The author also wishes to express his appreciation of the technical assistance given him by Messers. G. Middleton, R. Besel, and T. Dunton.

I. INTRODUCTION

The Navy's A-7 "Cutlass II" attack aircraft has had a recurring problem with the "hook point attachment bolt" which is a part of the aircraft's arresting hook mechanism. When landing on the flight deck of an aircraft carrier, the aircraft's arresting hook engages one of four arresting wires, called "cross deck pendants". The wires are attached to arresting engines beneath the flight deck which apply a constant decelerating force to the aircraft through the arresting wire and bring the aircraft to a stop. The decelerating force is transmitted from the wire through the hook point to the hook shank and then through the shank to the main aircraft structure.

The hook assembly and load path are illustrated in Figures 1 through 5. The design load path of the wire decelerating force is from the hook point to the shank through the large boss on the shank which mates with a receptacle in the hook point (Fig. 6,7). The attachment bolt was not designed to be a load carrying member and its sole purpose is to maintain a tight joint between the hook point and the boss of the shank.

The problem has been that the attachment bolt and nut have experienced a significant loss of torque, presumably as a result of arrested landings. When the point is only loosely held to the shank, it is possible for the dynamic loads of arrestment to move the hook point down, transferring

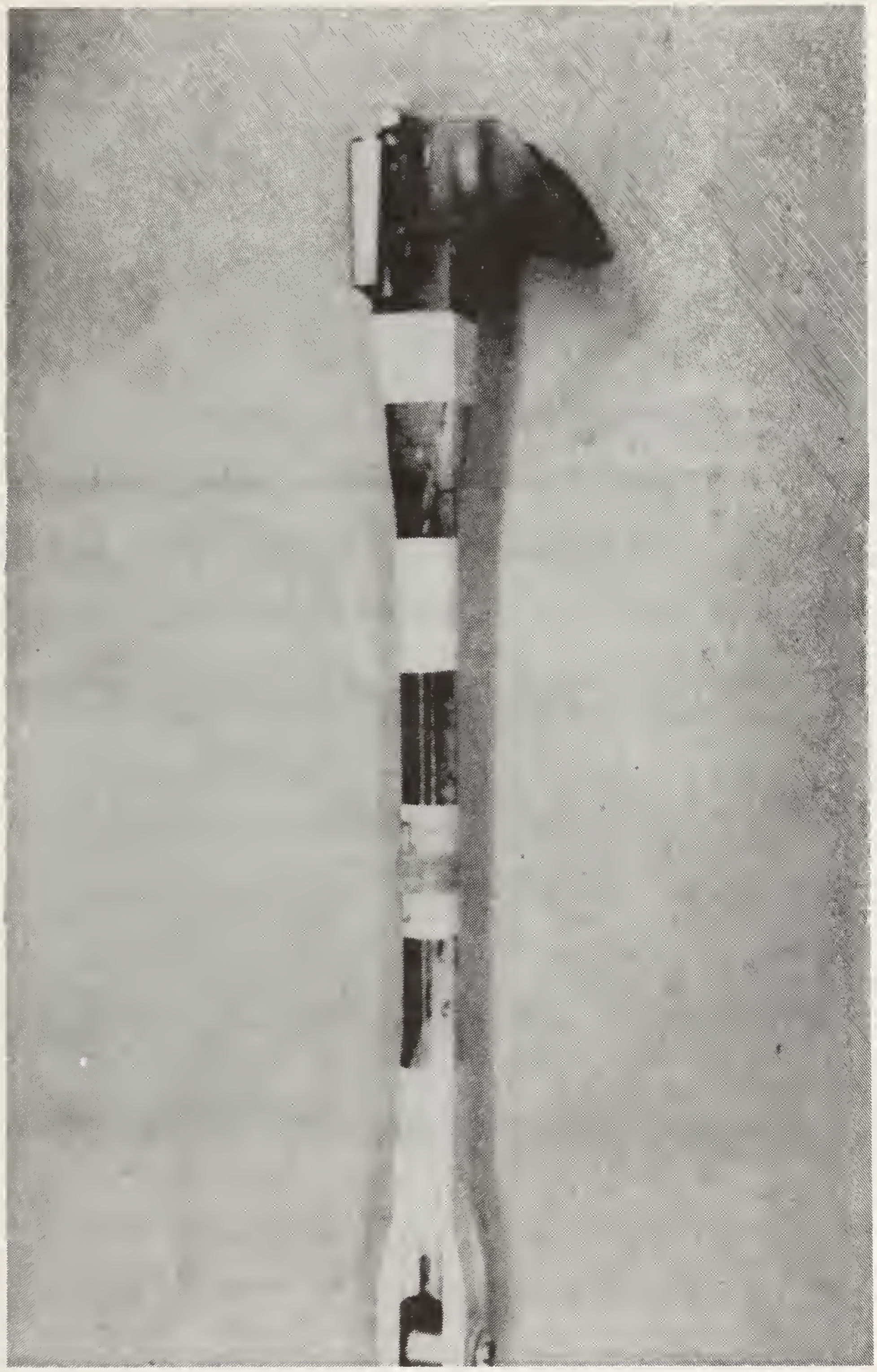

Figure 1. Complete arresting hook assembly.

Figure 2. Exploded view of hook point-boss joint.

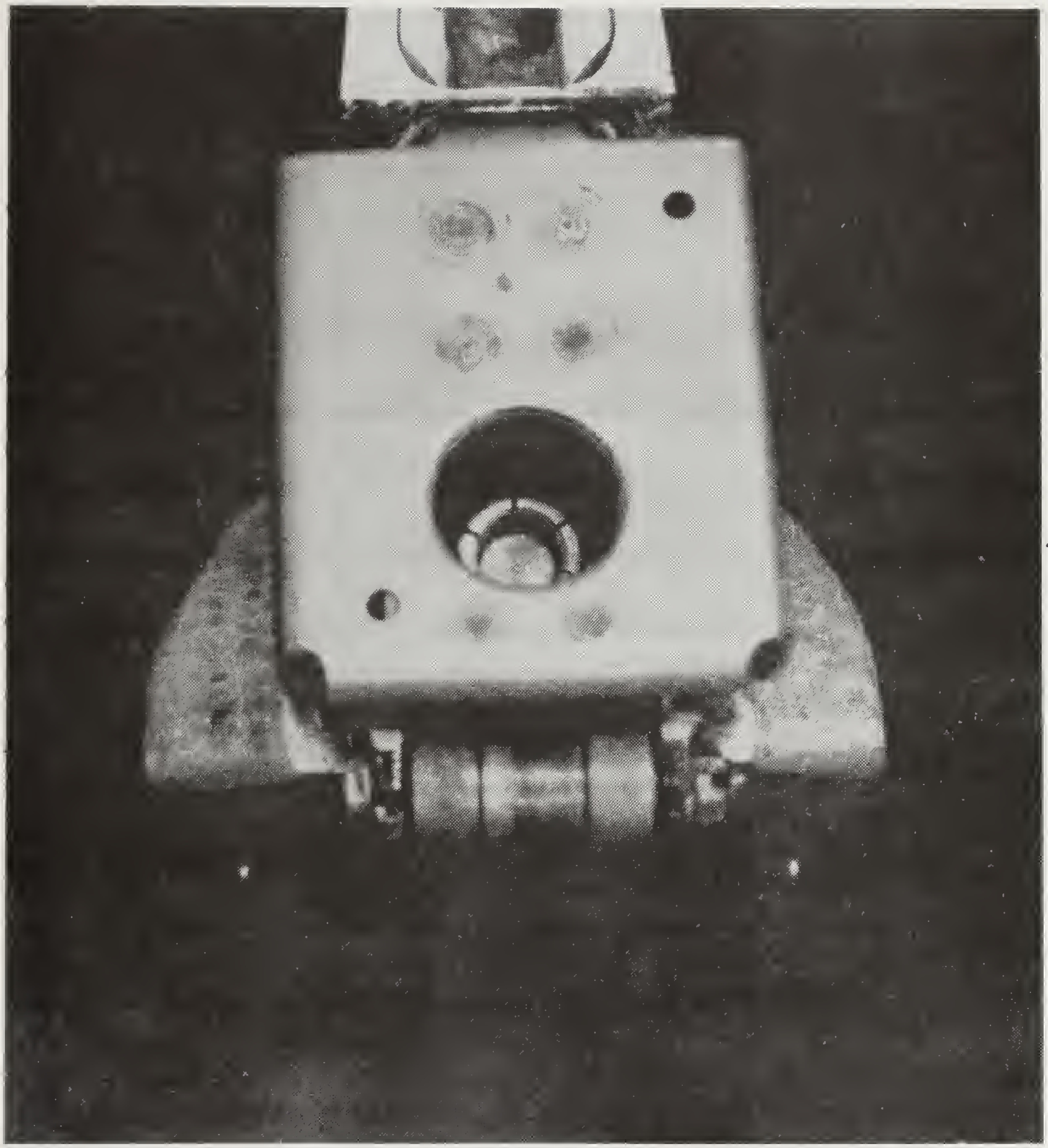

Figure 3. Top view of hook point-boss joint showing nut seated in the shank.

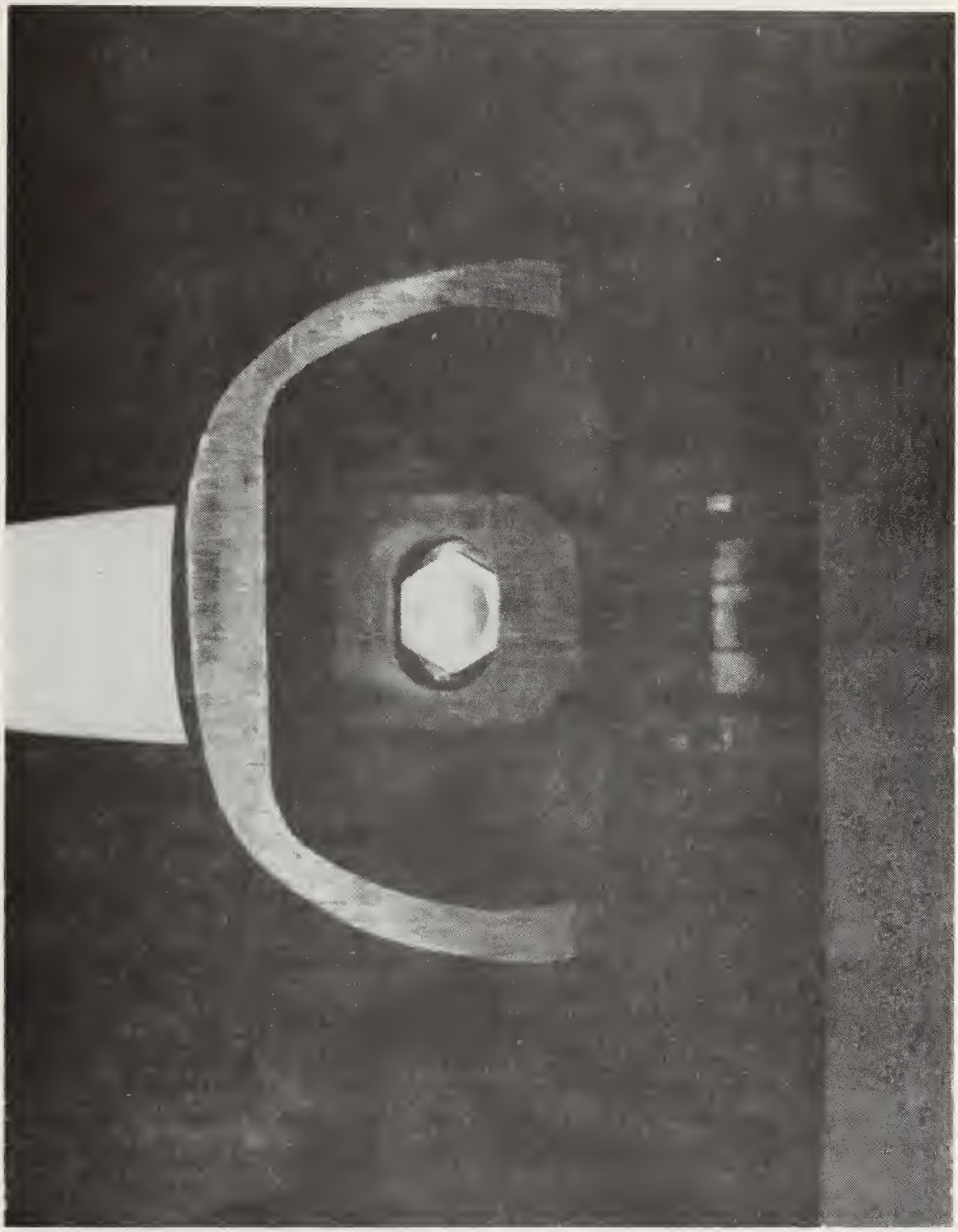

Figure 4. Bottom view of hook point-boss joint showing bolt head seated in the hook point.

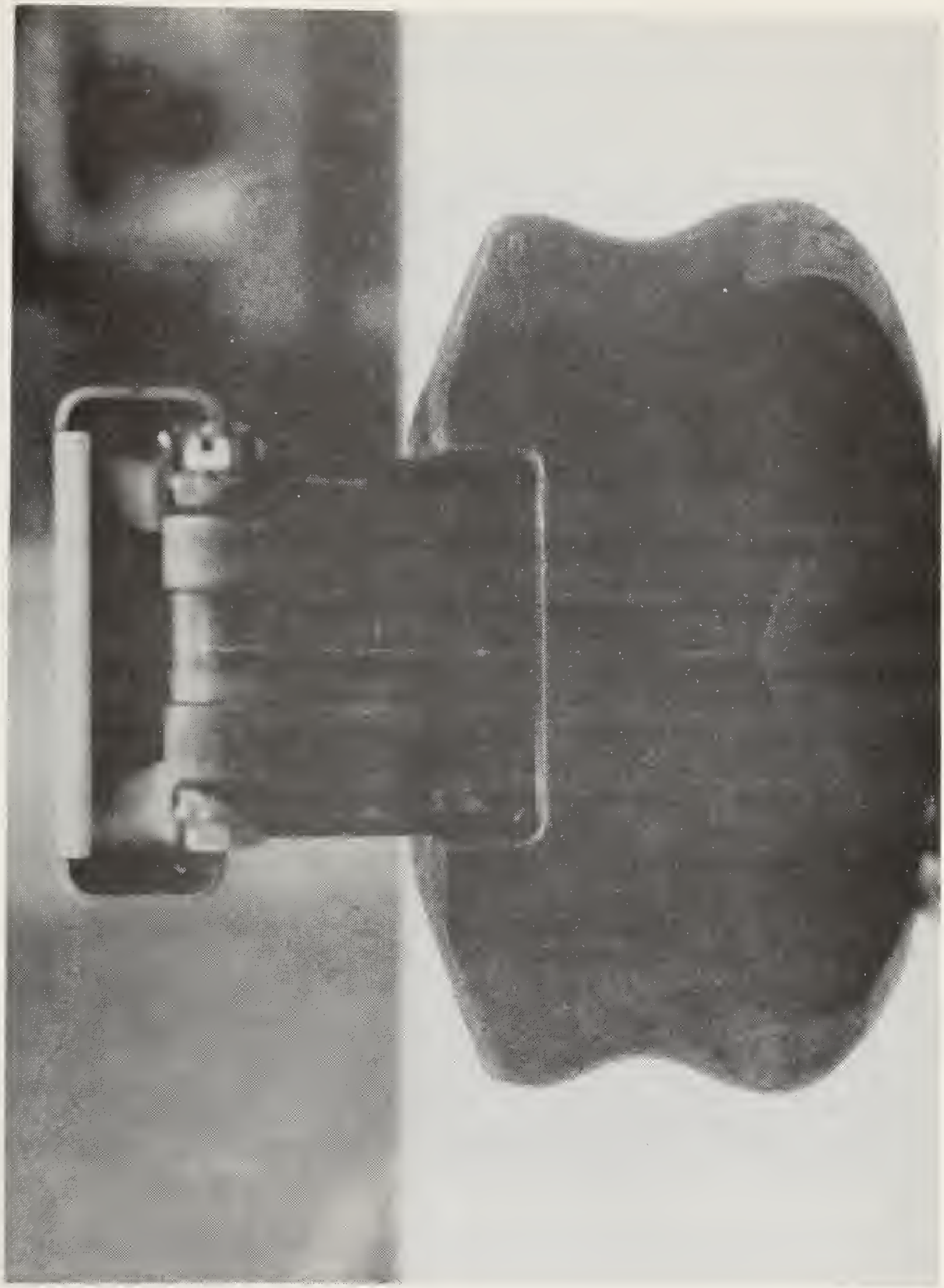

Figure 5. Back view of hook point-boss joint showing the hook point flanges fitted on each side of the shank to prevent rotation about the boss.

Figure 6. Side view of hook shank boss.

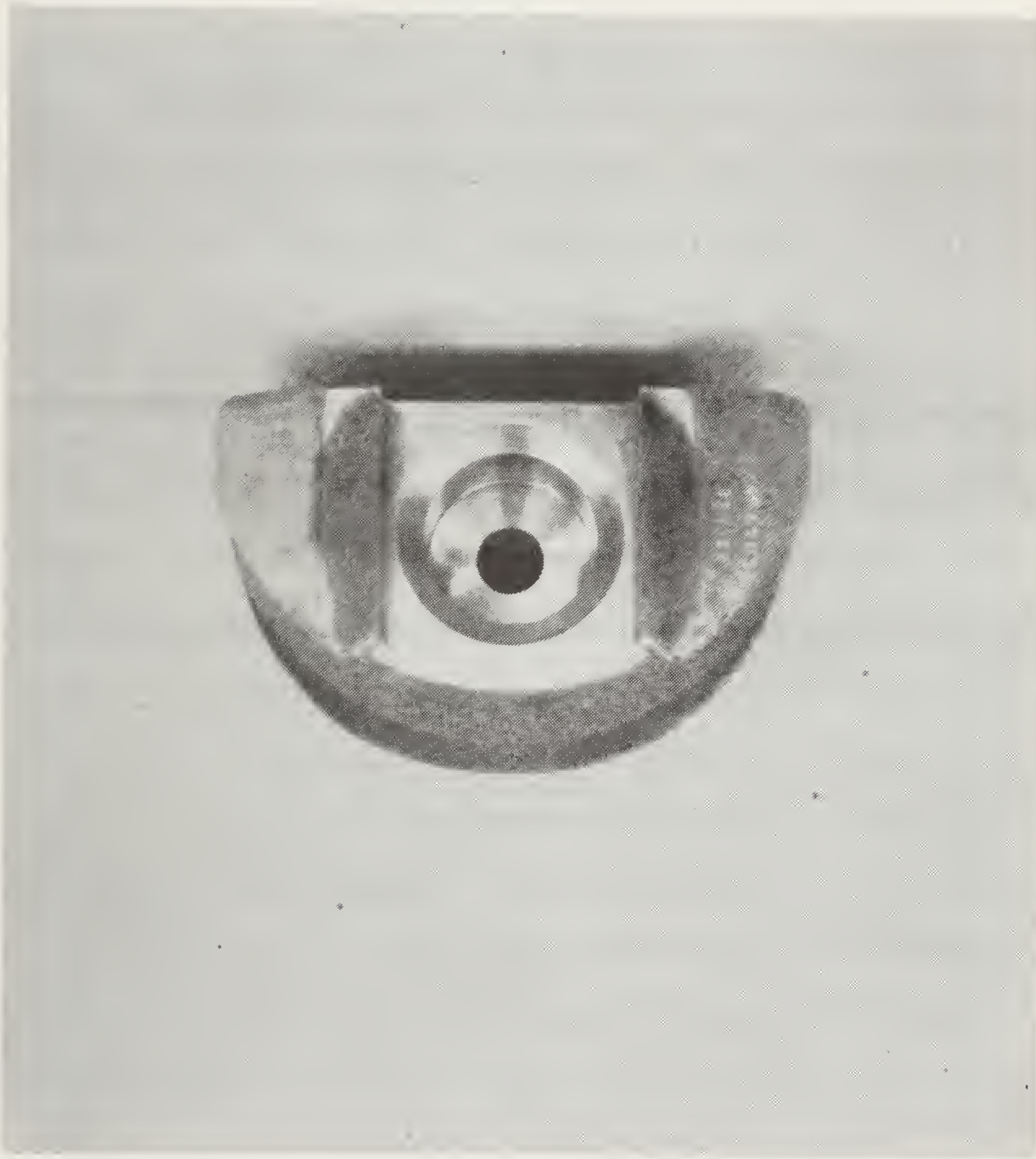

Figure 7. Top view of hook point showing the boss receptacle.

load from the boss to the bolt until the bolt is overstressed and the point separates from the shank. This chain of events is the probable cause for an incident which occurred in 1972 board the U. S. S. Kitty Hawk (CVA 63)(Ref. 1). An A-7 of Attack Squadron 195 (VA 195) made an apparently normal landing, the hook engaged the number three-wire (the target wire on Kitty Hawk), and began the arrestment. After fifty feet of wire runout, the hook point separated from the aircraft. Fortunately the aircraft had enough speed remaining to permit the pilot to fly the airplane off the angled deck, and a normal runway landing was made at a shore base. The hook point was left on the carrier deck, but the bolt and nut were not recovered.

Following this incident an investigation was begun and is continuing at this time to determine the cause and a possible "fix" to eliminate the problem. This thesis reviews the A-7 hook point attachment bolt problem and what has been done to try to correct the problem. It also reports on the investigation of another possible cause factor not previously considered, which is the self-loosening of threaded fasteners.

II. BACKGROUND

There have always been hook failures where a hook would hit a carrier deck obstruction and be torn off, a manufacturing defect would cause the hook to separate upon wire engagement, or improper installation would have a similar result, but coincident with the incorporation of a new style hook point on the A-7 arresting gear, a significant increase in hook failures occurred (Ref. 2). The newer "shovel nose" hook point was installed in order to reduce the high number of aircraft bolters, which occur when the hook misses all the wires and the aircraft has to take off again, due to the hook skipping over the arresting wires. The face of the shovel nose hook point makes a positive acute angle with the deck (Fig. 8), which forces the arresting wire into the hook point cable groove. The older "blunt nose" hook point, with the lower part of the hook face below the cable groove sloping away from the plane of the hook point face (Fig. 9), made arrestment very improbable unless the wire was engaged directly in the hook point groove or above.

The original design assumption of only the preload torque of 350 lb-ft. ± 25 (Ref. 3) and hook point weight forces being carried by the attachment bolt was essentially correct for the "blunt nose" hook point. The blunt nose has a tendency to bounce off deck obstructions due to its face geometry and the only place on the face which could be caught is in the cable

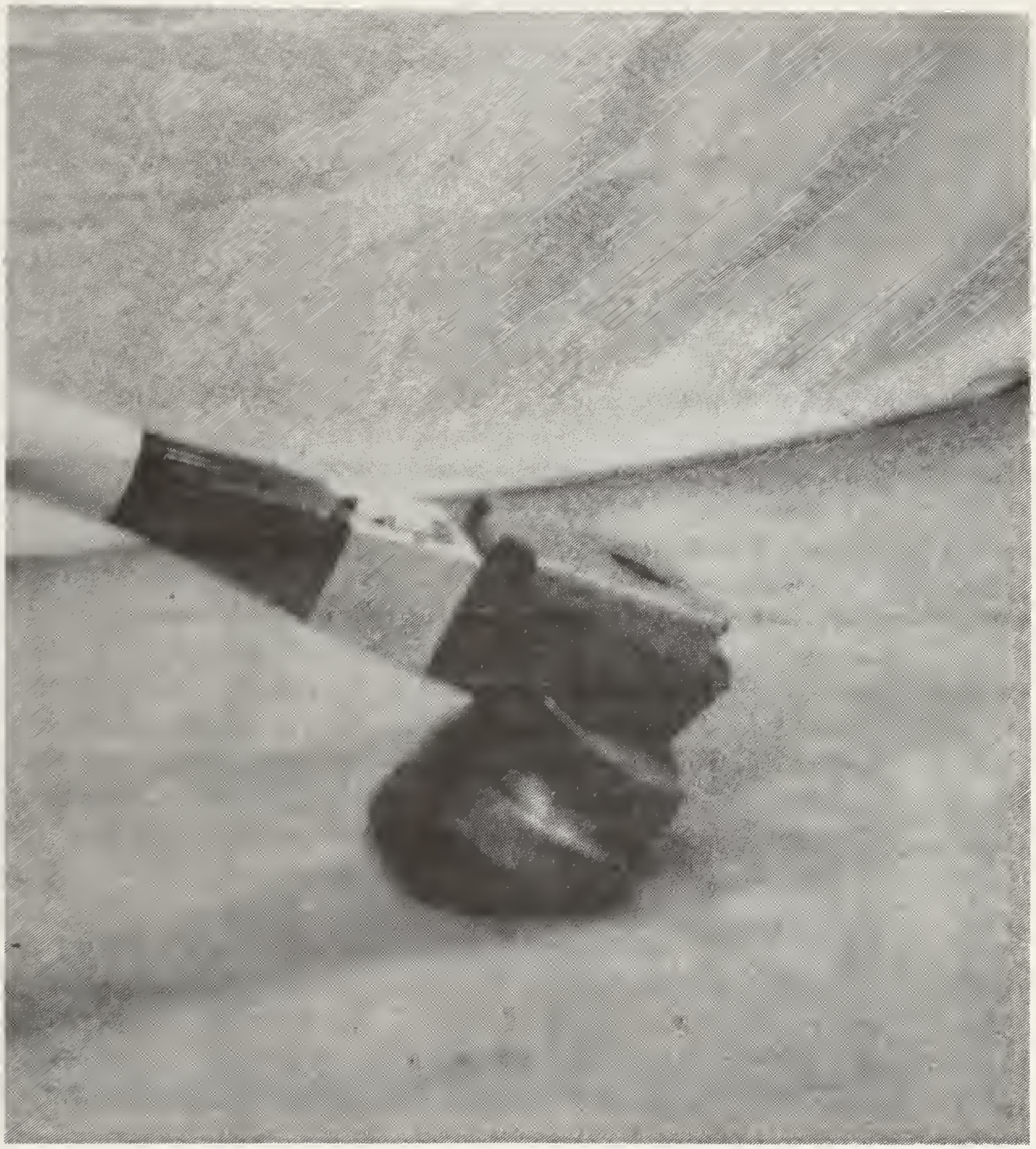

Figure 8. Actual attitude of arresting hook before wire engagement.

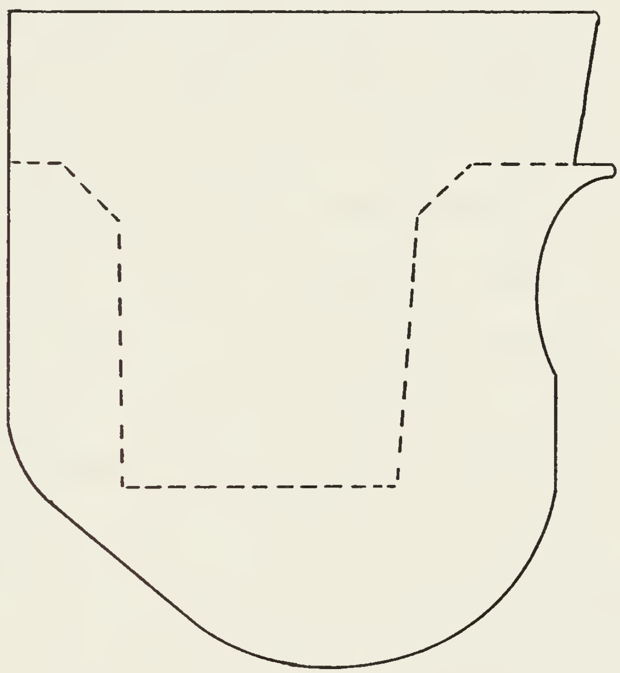

Figure 9. Old "blunt nose" hook point.

groove which, due to its position relative to the shank boss, transfers all of the impact loads to the shank boss (Fig. 7). In contrast the "shovel nose" point, due to its geometry (Fig. 10), has more of an opportunity to hang up on the edge of the nose where loads other than design loads could be imposed on the attachment bolt (Ref. 4). The relative position between the shank boss and front edge of the new hook point (Fig. 10) is such that it allows the possibility of significant rotational moments and tensile loads as well as shear to be applied to the hook point bolt-nut joint (Ref. 4,5) during arrestment or impacting of deck obstructions. Naval Air Rework Facility, North Island, during an engineering investigation, requested in Reference 6, of an A-7 attachment bolt failure reported that separation was initiated by the imposition of a fore and aft rotational moment which induced tensile, bending, and shear loading on the attachment bolt (Ref. 7). Any excessive clearance between shank boss and hook point receptacle or looseness in the joint greatly enhances the possibility of failure by allowing even more of the impact load to be imposed on the bolt.

The hook failure problem had been with the A-7 aircraft for a long period of time (Ref. 8) with one of the earliest reported in May 1970 on the U. S. S. Oriskany (Ref. 9), but the loosening problem did not begin to be recognized until the failure in March 1972 aboard the U. S. S. Kitty Hawk (Ref. 1). After the failure, VA-195 checked and replaced the attachment bolts on the remaining aircraft, and in doing

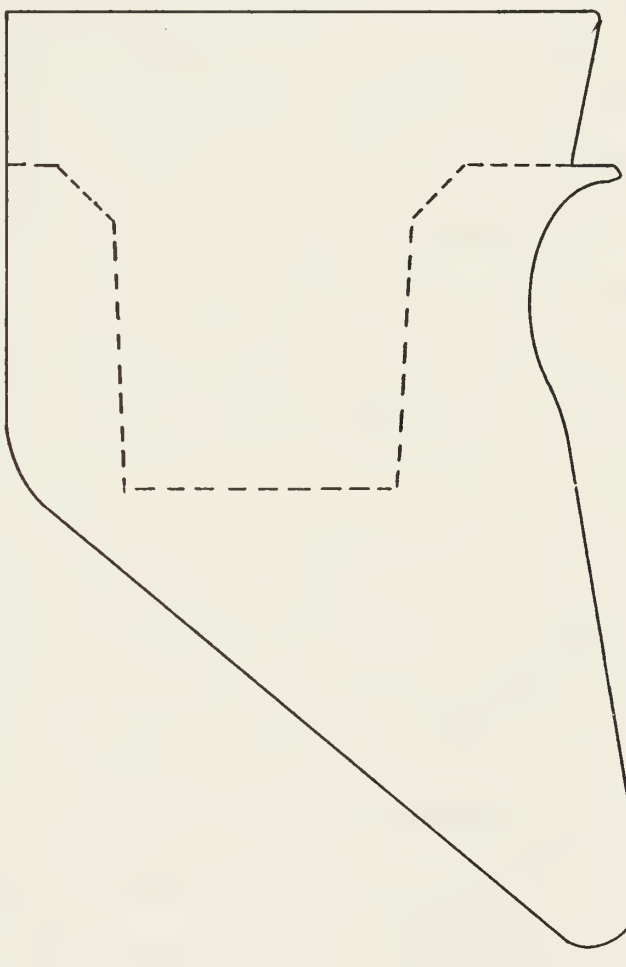

Figure 10. New "shovel nose" hook point.

so found one new bolt-nut combination which could only be torqued to 200 lb-ft. before nut slippage occurred. Upon checking the bolt, the squadron found that the pitch diameter was .03 inches undersized at the grip end of the bolt threads for several threads then tapered to the required size which was maintained for the last half of the thread length. The pitch at the necked down area also varied from the required .0625 (16 threads per inch) and the bolt was over the max length by .039 inches (Ref. 10,11). The squadron checked the other forty bolts the ship had on hand and found one more bolt with the same discrepancy. VA-94 had a hook point failure in which the nut was pulled over the top of the bolt threads with some of the bolt threads remaining in the nut threads (Ref. 12).

These reports started an inspection of the existing inventory of attachment bolts and nuts throughout the supply system which finally ended in NAVAIRSYSCOM requiring NARF Jacksonville, on the east coast, and NARF North Island, on the west coast, to inspect all attachment bolts and nuts (Ref. 13). As an example of the magnitude of the bolt-nut irregularity problem NARF North Island had to reject 795 bolts out of 2300 inspected (Ref. 14). The inspected nut and bolt combinations were to be packaged in matched pairs and a green dot to be put on the pair. Squadrons were only to use green dot bolts and nuts (Ref. 15,16).

The problem reappeared when, in first part of calendar year 1973, the hook points with the green dot, the rigidly

inspected matched bolt-nut sets, started to have failures. VA-125 had an A-7 lose a hook point on an arrested landing and in response checked the torque of their remaining aircraft (Ref. 17). They found two out of seven aircraft had under-torqued hook points, one plane had only 50 lb-ft. of torque preload after a total of eight arrested landings, and another had 250 lb-ft. after two landings. These findings prompted VA-125 to test other aircraft during carrier qualifications to find out if loosening was occurring during arrested landings or if maintenance personnel were not correctly torquing the bolts (Ref. 18,19). On 17 February 1973 they took five aircraft and carefully replaced the attachment bolts and torqued them to the correct values. During carrier qualifications on 18 February they monitored the torque after each arrested landing ("trap") on three aircraft and after every other landing on two aircraft. The results were that of the three aircraft being monitored after every landing, one aircraft lost torque to 290 lb-ft. on the eighth trap; and of the two being monitored every other trap both lost torque, one to 50 lb-ft. on the sixth trap and the other to 280 lb-ft. torque on the eighth landing. This proved that the loosening was not due to incorrect maintenance procedure but to a loosening mechanism in the hook point attachment bolt-nut combination. Also the rigid inspection by the NARFs ruled out improperly mated bolt-nut combinations. [Ref. 20]

In April 1973 NAVAIRSYSCOM HQ requested contractors to ship eight new high strength bolts made of H-11 steel and heat-

treated to 220 KSI and eight nuts made of steel and heat-treated to 180 KSI for test and evaluation (Ref. 21,22). In the meantime the squardrons are using the green dot matched sets until the new bolt and nut are approved (Ref. 23), and are directed to torque the bolts after each arrested landing (Ref. 24,25). These procedures still remain in effect today.

III. LOOSENING MECHANISMS

A. SELF LOOSENING

The first step necessary in formulating the theory of self loosening of threaded fasteners is the description of the load transfer between the nut and bolt in a static situation. J. N. Goodier (Ref. 25) was the first to describe nut loads and deformation under load conditions. Later, M. Heteyni (Ref. 26) performed photoelastic studies of threads under loads which supported Goodier's earlier experiments. These early studies led directly to the explanation of a self loosening theory for threaded fasteners by J. N. Goodier and R. S. Sweeney (Ref. 27,28).

The nut and bolt assembly showing the helix angle α, pitch diameter r, effective base radius r_1, and the thread form angle β is presented in Fig. 11. In order to describe the forces exerted on the threads of the bolt in a loaded condition, a coordinate axis transformation is necessary. The initial coordinate system with one axis in the bolt axial direction, unit vector $\hat{j}$, the second axis in the radial direction, unit vector $\hat{i}$, and the third axis in the circumferential direction, positive in the tightening sense, unit vector $\hat{k}$, must be transformed into a system with unit vectors $\hat{i}''$, $\hat{j}''$, $\hat{k}''$, where the $\hat{j}''$ axis is normal to the thread surface, the $\hat{i}''$ axis is tangent to the thread surface in the radial direction, and the $\hat{k}''$ axis is tangent to the thread surface in the

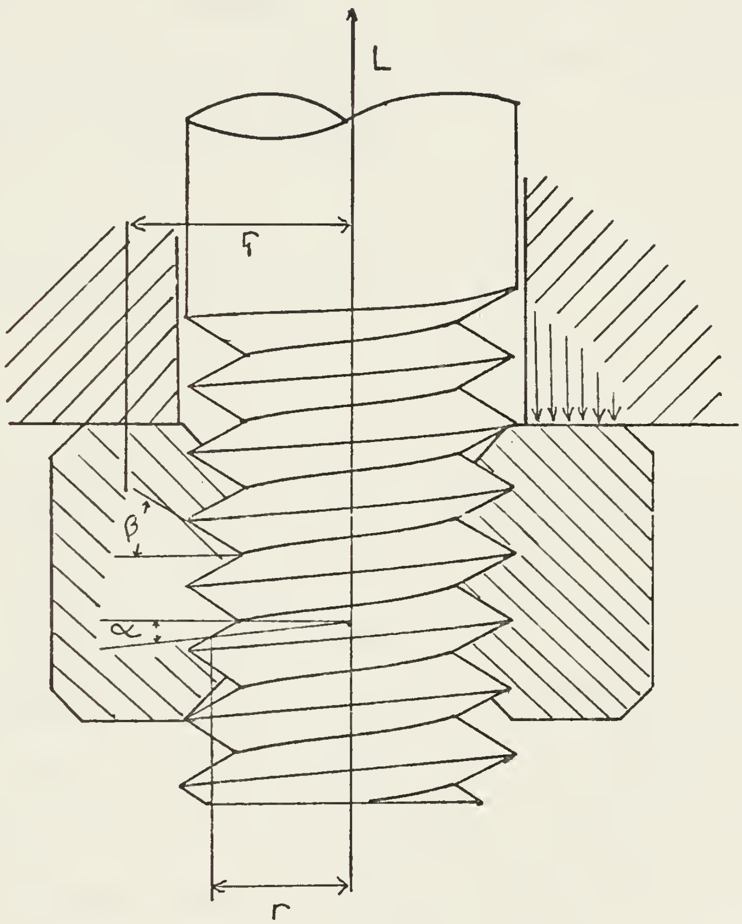

Figure 11. Parameters used in the nut and bolt assembly.

circumferential direction with same positive sense as the initial system. To accomplish the transformation, two rotations of axes are needed: the first rotation is counterclockwise about the $\hat{i}$ axis through the helix angle α resulting in the new system $\hat{i}'$, $\hat{j}'$, $\hat{k}'$. The second rotation is clockwise about the new $\hat{k}'$ axis through the thread form angle β which gives the following transformation equations from the $\hat{i}$, $\hat{j}$, $\hat{k}$ system to the $\hat{i}''$, $\hat{j}''$, $\hat{k}''$ system:

$$i'' = \hat{i} \cos\beta - \hat{j} \cos\alpha \sin\beta + \hat{k} \sin\alpha \sin\beta$$
$$j'' = \hat{i} \sin\beta + \hat{j} \cos\alpha \cos\beta + \hat{k} \sin\alpha \cos\beta$$
$$k'' = - \hat{j} \sin\alpha + \hat{k} \cos\alpha$$

Now the thread loads can be written in terms of the bolt loads. Of special interest is the sign of axial bolt-load component in the circumferential direction. Recalling that the positive circumferential direction in the new thread oriented coordinate system is in the tightening sense, a bolt under axial tension only is seen to have a negative circumferential force applied to its threads. Thus the bolt under tension will always have a component force of magnitude $F_o = L \sin\alpha$ in the loosening sense. This loosening force results in a loosening torque which is approximated as $T_o \approx Lr \tan\alpha$ (Ref. 28,29).

The loosening torque in the static situation is resisted by the friction torques produced between the bolt and nut threads, T_{FT}, and between the base of the nut and its seat, T_{FB}. The friction torques are approximated by $T_{FB} = Lr'\mu_B$

and $T_{FT} = Lr\mu_T$ (Ref. 29). In order to have the bolt-nut combination self-loosen, the loosening torque must overcome the combined effect of both friction torques, T_{FT} and T_{FB}. R. J. Sweeney (Ref. 28) using approximate formulas calculated that a 1/4"-20 coarse national thread bolt-nut combination has friction torques totalling approximately 5.7 times the loosening torque, and a 1"-8 national coarse thread combination has friction torques on the order of 8.8 times the loosening torque. Thus in the static case threaded fasteners are self-locking with no chance for self-loosening to occur.

The essential element of the self loosening mechanism is the motion of the nut when the bolt is subjected to a load. Initially the transfer of the applied load to the nut is concentrated near the base with relatively small pressures near the free end of the nut (Ref. 25,28). This action of the nut causes the thread bearing pressure to be concentrated at the base end of the nut during initial loading (Ref. 26,28). Typically the bolt is in tension and the nut is in compression, and as the load on the bolt increases, the bolt tends to contract radially due to the Poisson effect and the nut being further compressed tends to expand radially due to both the Poisson effect, and the thread form angle β. The radial component of the bolt load on the nut due to β tends to expand the nut wall radially, and as the nut expands, the threads at the base are deflected as cantilever beams (Fig. 12) causing further expansion of the nut. The expansion of the nut is greater at the base than at the free end as is shown in Fig. 13,

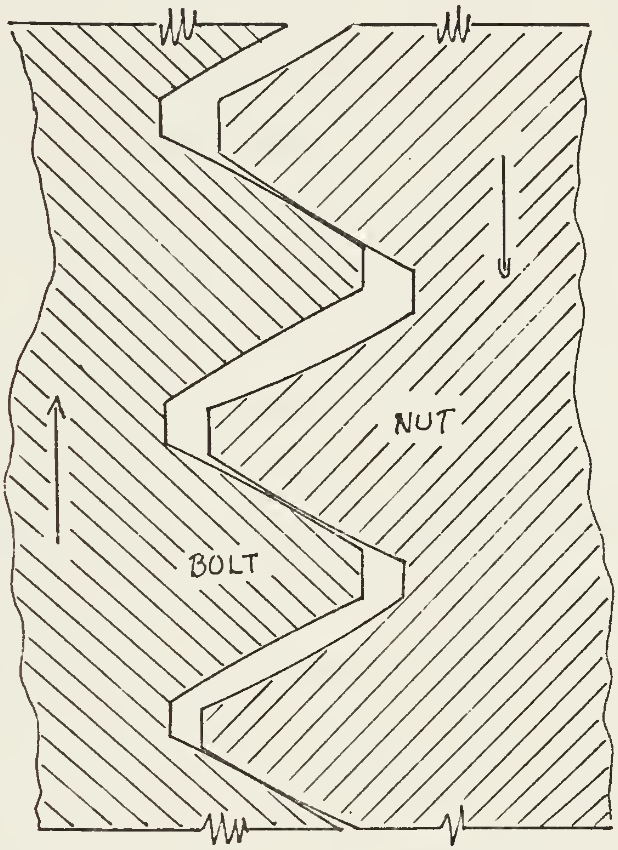

Figure 12. Deflection of the bolt and nut threads under a load.

and at higher loadings the thread bearing pressure is more evenly distributed throughout the nut length due to the uneven expansion. The distortion of the nut tends to concentrate the base reactions near the inner radius of an initially flat base due to the upward pull of the threads (Fig. 13) and the downward reaction of the base acting on different radii, and this results in a reduction in the effective base friction radius.

Now consider the above motion of the nut when subjected to a dynamic load situation. First consider axial tension only. As the load is applied, there is relative motion between the threads as the nut expands and the bolt contracts, and there is also relative motion between the base of the nut and its seat. The motion is just reversed when the load is relaxed. As the loading is continually varied, there is constant relative motion between the areas mentioned above. It is this radial motion between surfaces that effectively lets the bolt-nut combination become free of friction in the circumferential direction. This effect, G. H. Junker (Ref. 29) points out, results from the physical effects of friction on two interacting solid bodies, of which he states, "As soon as the friction force between two solid bodies is overcome by an external force working in one direction, an additional movement in any other direction can be caused by the action of forces that can be essentially smaller than the friction force".

A simple example of the above physical law is a block on an inclined plane. The weight of the block in Fig. 14 is not

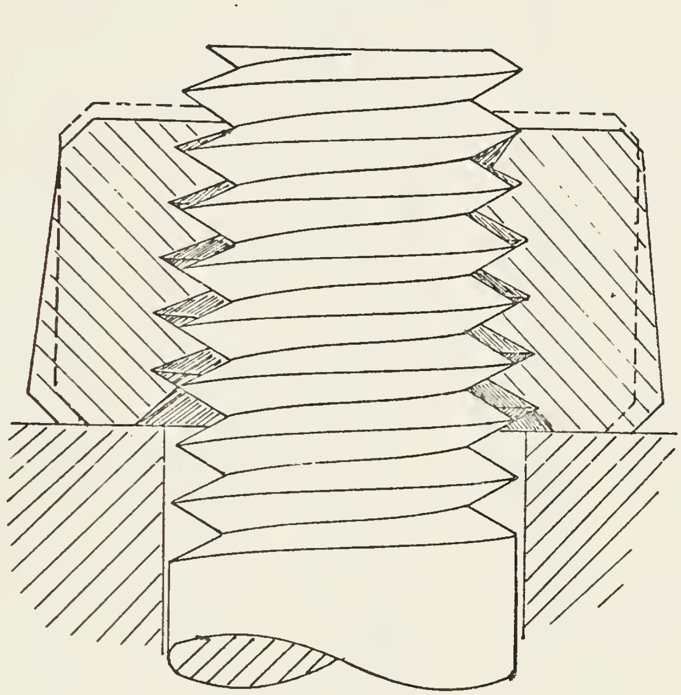

Figure 13. Nut under a load showing its radial expansion.

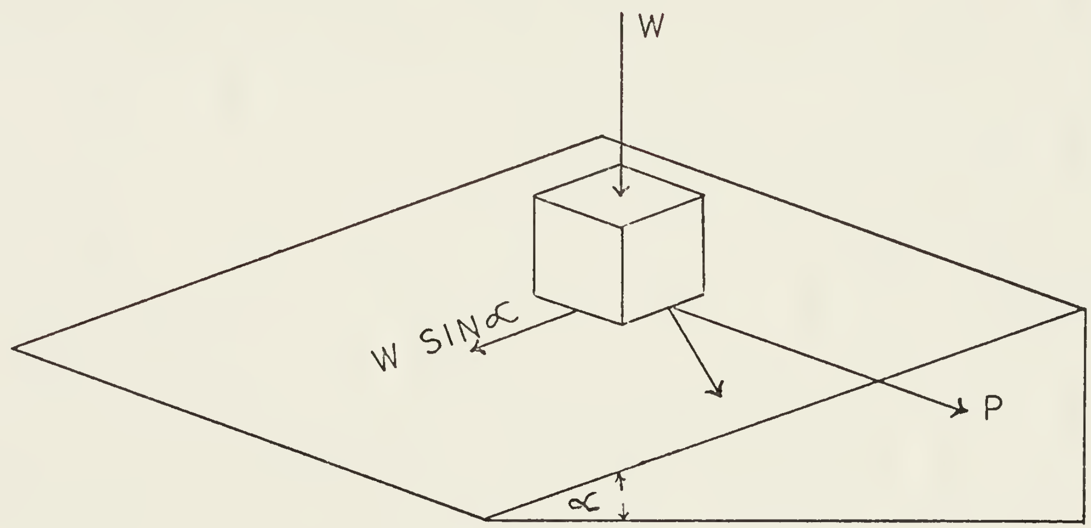

Figure 14. Motion of a block on an inclined plane with only gravity and a side load acting on it.

sufficient to overcome the friction force unless the component of weight parallel to the inclined plane $F_W = W \sin \alpha$ is greater than the friction force, $F_F = W \cos\alpha\ \mu$, where W is the weight, α is the angle of the inclined plane, and μ is the coefficient of friction. Now if a side force, P, is introcuded and is large enough to overcome the friction force and cause the block to move in the sideways direction, the block will not only move sideways but will also move down the inclined plane as indicated in Fig. 14. The downwards motion is due to the component of the weight that was initially too small to produce motion. An analogous effect takes place in a threaded fastener when cyclic loading or vibration has created relative motion in the bolt-nut joint. The loosening torque which was initially too small to overcome the friction torque now is acting on an essentially frictionless surface and will cause the nut to rotate in the off direction, and self loosening occurs. Total self loosening occurs when the nut rotates enough to release all of the torque preload previously applied and partial self loosening results when some of the preload is released but some still remains.

R. J. Sweeney and J. N. Goodier (Ref. 27) produced partial self loosening during dynamic axial load tests conducted with a 3/4"-10 national coarse thread bolt-nut combination. In 500 cycles a nut rotation of 5.5×10^{-3} revolution, approximately 2 degrees, was obtained. J. A. Sauer, D. C. Lemmon, and E. K. Lynn (Ref. 30) used a fatigue testing machine to apply cyclic axial tension loads to a 5/16"-18 national coarse

thread bolt-nut combinations to verify Goodier and Sweeney's theory. They used various dynamic-to-static load ratios ranging from .25 to 1.00, and found that generally as the dynamic-to-static load ratio increases, i.e. as the dynamically applied load approaches the static preload, the amount of loosening increases. The maximum recorded rotation was 6 degrees in 25,000 cycles with a dynamic to static load ratio of .8, and the maximum static preload relaxation of approximately 56 percent with a dynamic to static load ratio of 1.0. Thus again only partial self loosening was attained.

The first evidence of total self loosening was found in a study by S. K. Clark and J. J. Cook (Ref. 31) in which oscillatory vibrations applied to a bolted connection were used to produce bolt loosening. Two fixtures were bolted together with one acting as the nut and this one was oscillated with respect to the other. In this manner a relative rotational motion was produced between the two fixtures and loosening of the bolt occurred. Loosening curves were drawn for different type bolts with one type per graph, and each curve graphed the number of cycles until total loosening occurred versus torque amplitude for a given bolt preload. For example a 3/8"-16 place bolt with a tensile preload of 10 KSI took approximately 45 cycles at an oscillatory torque amplitude of approximately 150 lb-in. to toally loosen the fittings. The curves show that as preloads are increased, the number of cycles and/or the torque amplitudes increase to cause loosening.

G. H. Junker (Ref. 29) states that the most severe condition for the self lossening of bolt-nut combinations is that of transverse vibrations. When transverse loadings are applied, that is loadings that tend to make the bolt move perpendicular to its centroidal axis, and exceed the friction force between the clamped parts, transverse slippage results between the clamped parts. This transverse slippage forces the bolt to assume a pendulum movement, a wobbling of the bolt in the bolt hole, which leads to relative motion in the bolt hole and in the threaded areas. The transversely induced relative motion can have much larger amplitudes than the axially induced motion, and unlike the axially loaded case, the relative motion between thread surfaces will occur in all parts of the nut threads and not just close to the base of the nut (Ref. 29). Thus when the transverse slippage of the bolt becomes large enough, slippage of the nut bearing surface will occur and the joint will become free of friction in the circumferential direction, thus allowing the off torque to rotate the nut and loosen the connection. Junker reports verification of this theory in work done by Schoellhammer in the Daimler-Benz Laboratories. Schoellhammer determined the external off-torque necessary to loosen various fasteners under various conditions, including self-locking types under dynamic transverse loadings. In these latter tests the frequency of the transverse loading was increased until a point was reached where the external off-torque required to loosen the fastener began to decrease, and finally the nut rotated fully loose without the necessity of

any external off-torque. For these tests Schoellhammer used a vibration machine, with the fastener to be tested clamping a plate against a block. Junker also set up a transverse vibration test machine and tested this theory. His tests showed that the self loosening mechanism in screws is independent of frequency. It simply depends upon the occurrance of relative motion between thread surfaces and the length of the motion in one cycle (Ref. 32,33). The one case where frequency does affect the loosening process is when the transverse forces causing the motion are inertia forces: then the forces are a function of the square of the frequency (Ref. 32,33).

It is this result of Junker's testing, i.e. loosening is independent of frequency in his transverse vibration tests, which suggested that the first situaiton in which to test the A-7 attachment bolt for self-loosening should be the transverse force application case.

B. PLASTIC STRAIN

The two friction torques, T_{FT} and T_{FB}, oppose the rotation of the nut which allows the nut to be tightened without loosening when the externally applied torque is released. Each of these torques, T_{FT} and T_{FB}, is directly proportional to the axial load L, and thus any action which reduces the axial load will also reduce the total friction force in the bolt-nut combination.

The bolt-nut combination derives its clamping force, F_c, from the linearly elastic behavior of the bolt material assuming that the clamped parts are essentially rigid. "Torquing the nut down" stretches the bolt which acts as a linear elastic spring with an equivalent spring constant, $K_b = \frac{E_b A_b}{l_b}$. This is true as long as the strain of the bolt remains elastic; i.e. as long as the stress-strain state is between point O and B in Fig.15, where B is the proportional limit.

If the torque is increased so that the bolt is strained plastically, say to point C, two things occur. First the spring "constant" becomes a variable, so that the axial force is no longer simply $L = F_c = K_b e_b$. Secondly when the load is released the stress-strain state will be found somewhere on line CF, a line from C parallel to the original elastic line rather than on line OABC (Ref. 34,35). The distance OF is the "permanent set" or "plastic strain" of the bolt.

The question is, how much of the clamping force due to the original torque preload is lost when the bolt is plastically strained? First assume the bolt has been stressed to point A

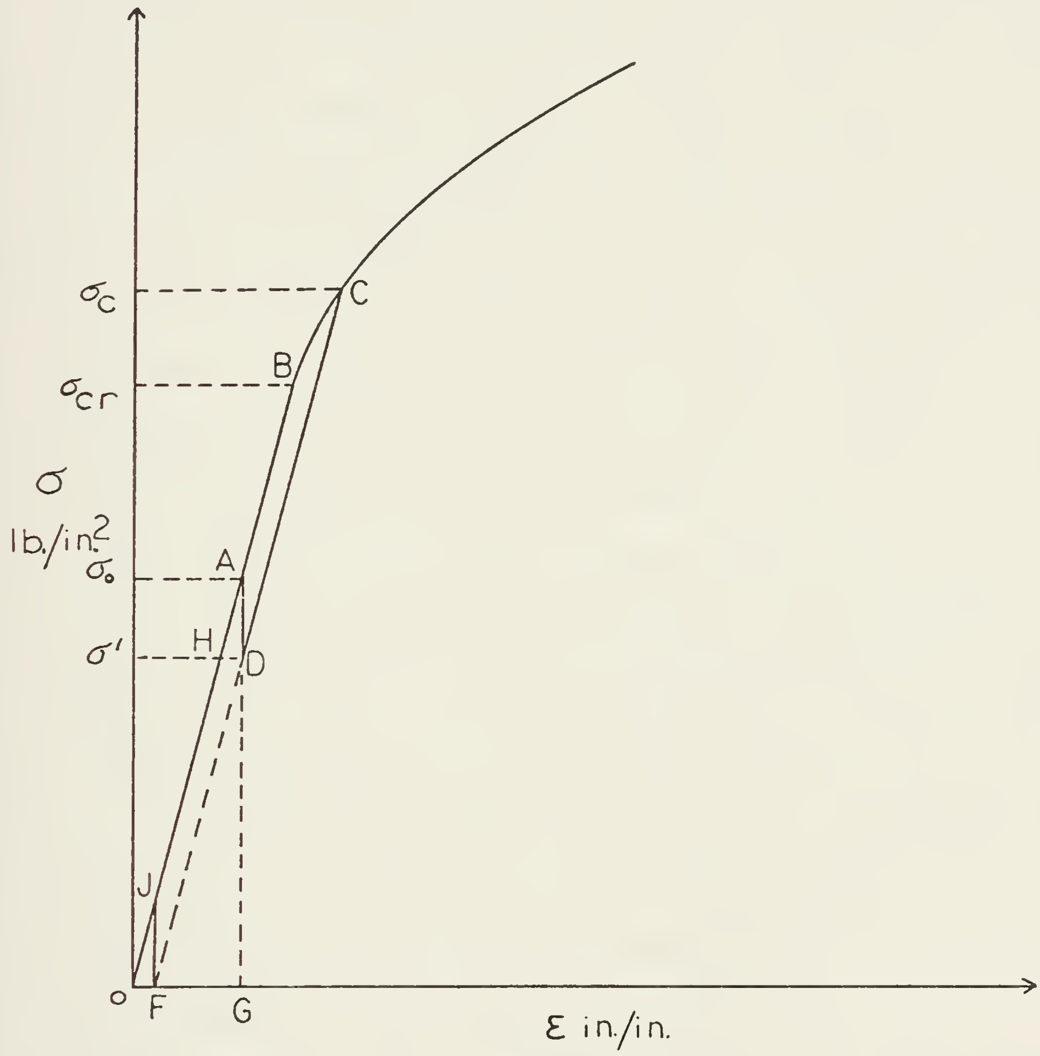

Figure 15. General stress strain diagram.

in Fig. 15 by the torque preload. Now some external force causes the stress to increase past B to C. When the external force is released, the stress will diminish along line CF, as described above. If it is still assumed that the clamped parts are essentially incompressible, the stress will decrease until the strain is equal to the initial strain OG in Fig. 15, i.e. until the distance between bolt head and nut is the same as it was after the initial torquing. Now, however, the total strain is composed of an elastic part, FG, and a plastic part, OF. The latter contributes nothing to the clamping force.

The stress-strain state has now returned to the elastic region via a line parallel to the elastic line as described above. Thus the unloading was accomplished along a line with the original Young's Modulus as the slope, and conforms to the relation, $G = E\varepsilon_b$, all of which implies that the original Young's Modulus can not only be used to describe the material's reaction to additional applied stress until C is reached again, but also that it can be used to calculate the loss in clamping force. Recalling the geometry of parallel lines, it is evident that line segments FJ and DA are of equal magnitude and both equal $E(\overline{OF}) = E(\overline{HD}) = E\varepsilon_p$. Thus the remaining clamping force, F_c, is equal to

$$L' = F'_c = A_b(\sigma - \Delta\sigma) = \frac{A_b E_b}{l_b}(\overline{OG} - \overline{OF}) = \frac{A_b E_b}{l_b}(e_b - \varepsilon_p) = \frac{A_b E_b}{l_b}(\varepsilon'_b)$$

using the relations for the friction torques given earlier, namely $T_{FB} = Lr_1\mu_B$ and $T_{FT} = Lr\mu_T$, an expression for both the

loss of torque, ΔT, and the remaining torque, T', can be derived:

$$T = T_{FT} + T_{FB} = L(r_1\mu_B + r\mu_T)$$

$$T' = L'(r_1\mu_B + r\mu_T) = \frac{A_b E_b}{l_b} (\varepsilon'_b)(r_1\mu_B + r\mu_T)$$

$$\Delta T = \Delta L(r_1\mu_B + r\mu_T) = \frac{A_b E_b}{l_b} (\varepsilon_p)(r_1\mu_B + r\mu_T)$$

IV. EXPERIMENTAL METHOD AND RESULTS

A. DESIGN OF TEST

The hook point is subjected to large applied forces not only from engaging the arresting wire or impacting deck obstructions but also from the dynamic response of the wire upon arrestment. Performance tests of the Mark 7 Mod 3 Recovery System (Ref. 36) indicate that aircraft which weigh less than 28,000 lbs. upon landing receive the largest hook loads during the "dynamic" region of the arrestment. The dynamic region is the interval of approximately .8 seconds from the initial hook contact with the wire until the transverse waves induced in the wire by the arrestment are damped out. In the case of the A-7 aircraft, these forces are on the order of 96,000 lbs. for a landing weight of 25,000 lbs. Any increase in landing weight gives a corresponding increase in hook loads.

The transverse wave front induced by the hook's initial contact with cable takes approximately .2 seconds to be reflected back to the hook point (Ref. 36). This allows the possibility of four force impulses during this dynamic interval of .8 seconds which the hook point would incur as the reflected wave front strikes the hook point. Junker's test result (Ref. 30) was that the frequency of the applied loading had little or no effect on the self loosening mechanism but that it was the ability of the load to cause relative motion between the parts of the bolted joint which was the determining factor for self

loosening to occur. With this in mind it seemed possible that the combination of the hook point hitting the deck and/or other obstructions together with dynamics of the wire engagement could cause the bolt to vibrate or "wobble" in the bolt hole, giving the conditions necessary for self loosening to occur.

Looking at the way the hook point attachment assembly is put together, Fig. 2, and how the bolt head is securely held in the hook point, Fig. 4, it can be seen that any sliding of the hood point relative to its seat on the shank results in transverse loading which Junker (Ref. 24) points out is the severest condition for the occurrance of self loosening. It was decided, based on the above, to investigate the possibility that large magnitude, low cycle, and low frequency hook loading of an arrested landing could cause the attachment bolt-nut combination to self loosen.

The first thing to determine was if there was any possibility of relative motion between the hook point and the shank boss. Measurements were taken on three operational A-7A aircraft of VA-125 at NAS Lemoore, California, and the results are given in Table 1. The aircraft engineering drawings (Ref. 37) call for a boss outer diameter of $1.902^{+.000}_{-.002}$ inches. In measuring the boss on the aircraft, it was discovered that the diameter at the free end, diameter A in Fig. 16, was smaller than the diameter closer to the shank, diameter B. Diameter A averaged 1.90067 inches and diameter B averaged 1.9013 inches. The hook point receptacle was also measured

TABLE I.

A-7A BOSS AND HOOK POINT MEASUREMENTS

Aircraft Bureau Number	Diameter A	Diameter B
Boss on 153173	1.9005	1.9015
Boss on 153151	1.901	1.9015
Boss on 153199	1.9005	1.901
New Hook Point	1.9015	1.9035

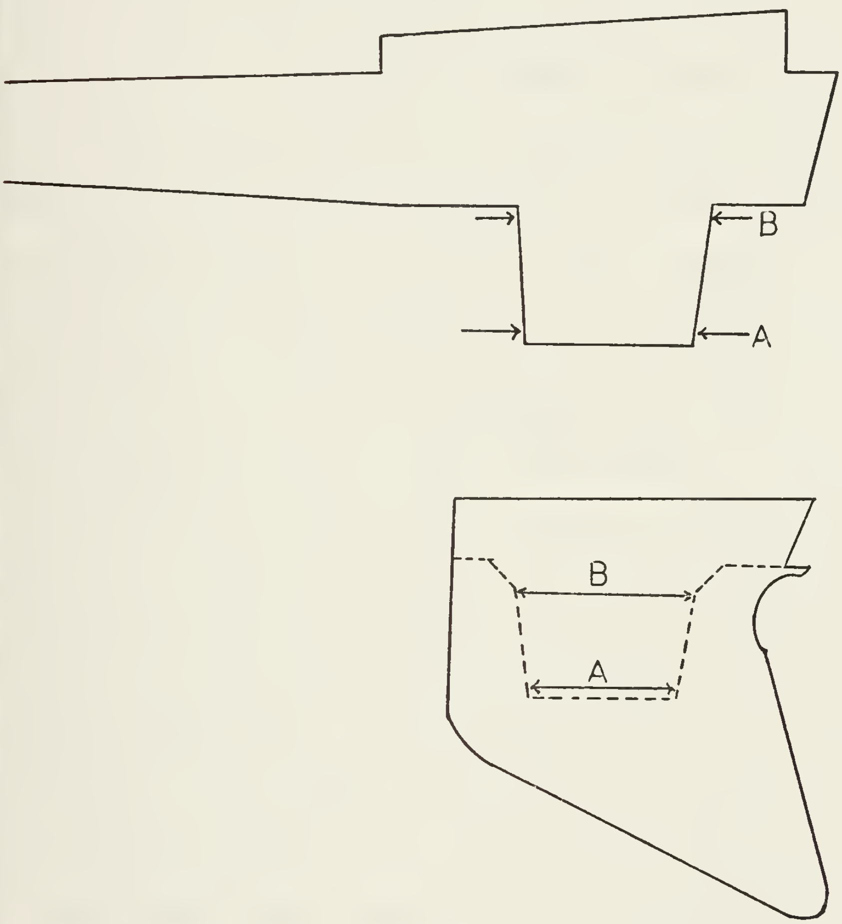

Figure 16. Positions measured on the boss and the hook point receptacle.

and the measurements of a new hook point were: Diameter A in Fig. 16, corresponding to diameter A of the shank, was 1.9015 inches and diameter B was 1.9035 inches. Upon inspection of the loosely mated parts, the hook point could be made to rock back and forth, pivoting about the end of the boss. The hook point could also be made to rotate slightly with the boss acting as its axis of rotation. These motions were accomplished by hand force alone with the bolt sitting loosely in its seat. Thus the possibility of relative motion between hook point and shank boss exists in the arrested landing.

The bolt diameter relative to the bolt hole diameter was measured also. The aircraft drawings (Ref. 37) specified the hole diameter to be $.7500^{+.002}_{-.000}$ inches. The minimum and maximum diameter of the bolt was .7406 and .7500 respectively. This gives a possible clearance of .0114 inches between the grip of the bolt and the bolt hole sides. The measured clearance between shank bolt hole diameter and bolt diameter were on the order of .003 inches. This does show that there is room for the bolt to vibrate or "wobble" in the bolt hole.

The next thing which needed to be checked was to see if the bolts were extending. Therefore twelve new bolts were requested from COMNAVAIRPAC in order that they be measured and then sent to VA-147 to be used in carrier qualifications. After ten arrested landings they were sent back for another measuring with the results in Table 2. It appeared at that time that the measurements showed insignificant permanent strain in some bolts and none in others. This occurance,

TABLE II

TEST FOR PLASTIC DEFORMATION DURING CARRIER LANDINGS. DATA ON TWELVE BOLTS, EACH SUBJECTED TO TEN ARRESTED LANDINGS.

Bolt No.	Length Before Test - Inches	Length After Test - Inches	Microstrain x 10^6 in/in.
1	4.2825	4.2828	70.05
2	4.2954	4.2982	651.8
3	4.2902	4.2903	23.3
4	4.284	4.284	0
5	4.2935	4.2935	0
6	4.292	4.292	0
7	4.291	4.2912	46.6
8	4.2922	4.2925	69.9
9	4.2825	4.2825	0
10	4.304	4.304	0
11	4.304	4.3045	116.2
12	4.295	-	-

therefore, pointed even more to a possible self loosening mechanism being present in the hook point - shank boss bolted joint. It should be noted that during these tests the bolts were retorqued after each landing as per Reference 24, and no torque loss was reported by the VA-147 crew.

Thus having found that at least some of the ingredients for a self loosening mechanism were present in A-7 hook point attachment system, it remained to be demonstrated that the mechanism existed and produced significant torque loss.

The most desireable way to do this seemed to be to make a test fixture which would simulate the actual hook point/shank attachment, and apply loads which simulated the load-time history of an arrested landing. Data on the load-time histories of actual landings were available (Ref. 36). For a time it appeared that a programmable closed-loop electrohydraulic test machine might also be available so that this plan would be feasible; unfortunately the machine never materialized.

The second choice was to use a mechanical universal test machine, manually controlled, and to apply the loads as rapidly as possible to the desired magnitude. There could be no simulation of the dynamic time history, but the number and intensity of loads experienced could be simulated.

B. TEST EQUIPMENT

1. Test Machine

The test machine used was a Riehle Model PS-300 universal screw power testing machine. The machine had a maximum

crosshead speed of 2 inches/min. and this greatly limited the rate of load application. The test machine is shown in Figures 17 and 18 with the test fixture in place. The machine has to be manually controlled with a set of buttons marked "load", "stop", and "unload". The operator would push the load button and machine would increase the load until the operator pushed the stop button. There was a problem in obtaining exact load values because of the inertia of the screws and crosshead tending to give a little increase of the load on the specimen after the stop button was pushed. This problem was accentuated by the fact that the maximum crosshead speed was used. The operator, therefore, had to stop the machine at a loading value below what was desired in anticipation of the load increase due to the momentum of the machine parts. Thus loading ranges were set up which gave a plus or minus 1,125 lbs. about a desired loading. In the tabulation just the maximum load limit and the minimum load limit are given.

2. Bolt Length Gauge

The length measurements were taken with a Hewlett-Packard Linear Variable Differential Transformer. The linear range of .25 inches was graphed to relate millivolts to inches. A United Systems Corp. D. C. voltmeter was used with a 1000 millivolt scale. Figure 19 shows the measurement setup.

3. Torque Wrench

The torque measurements were taken with a torque wrench with maximum torque reading capability of 600 ft-lbs. The reading could be made to plus or minus 5 ft-lbs.

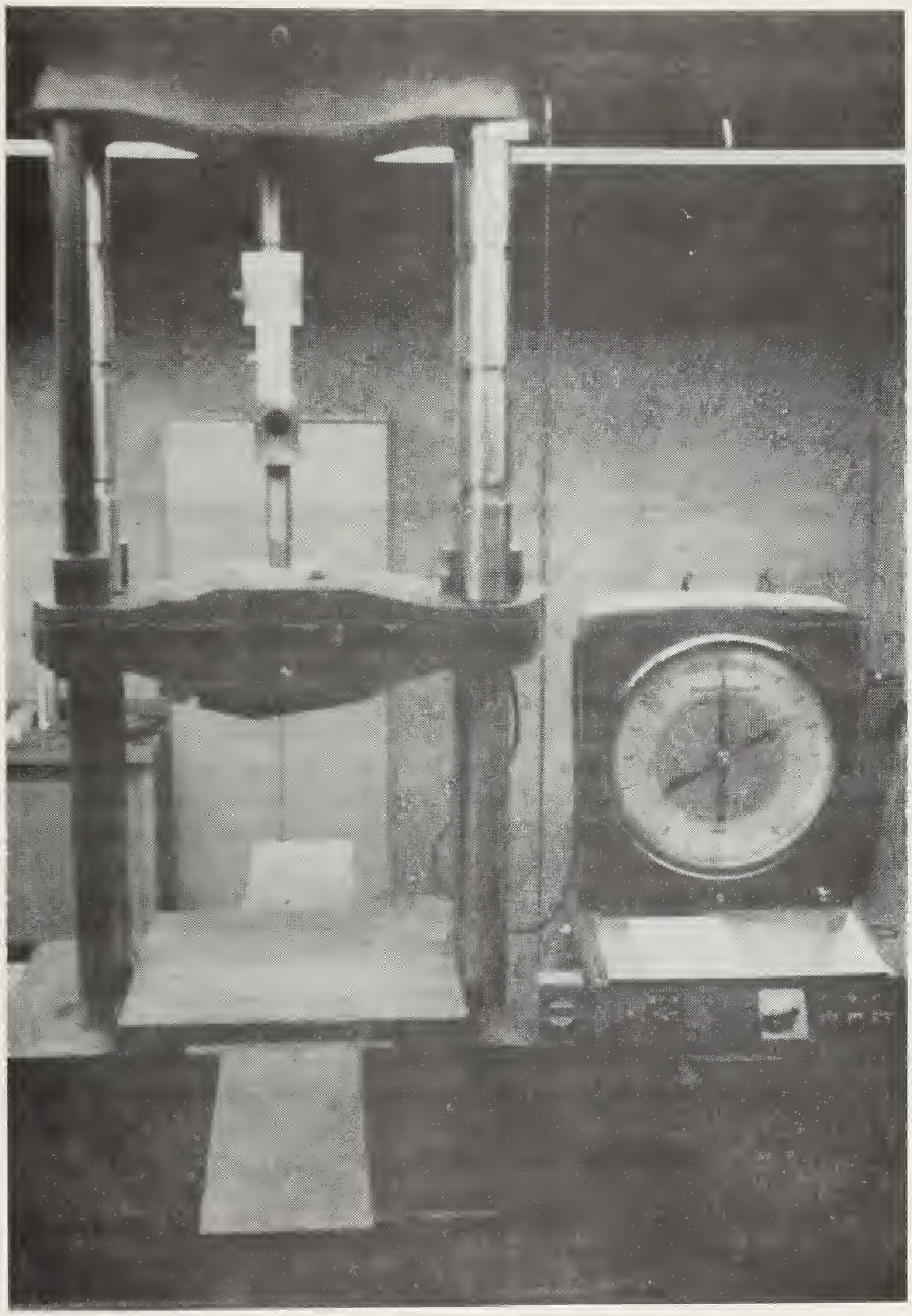

Figure 17. Riehle model PS-300 test machine with test fixture in place.

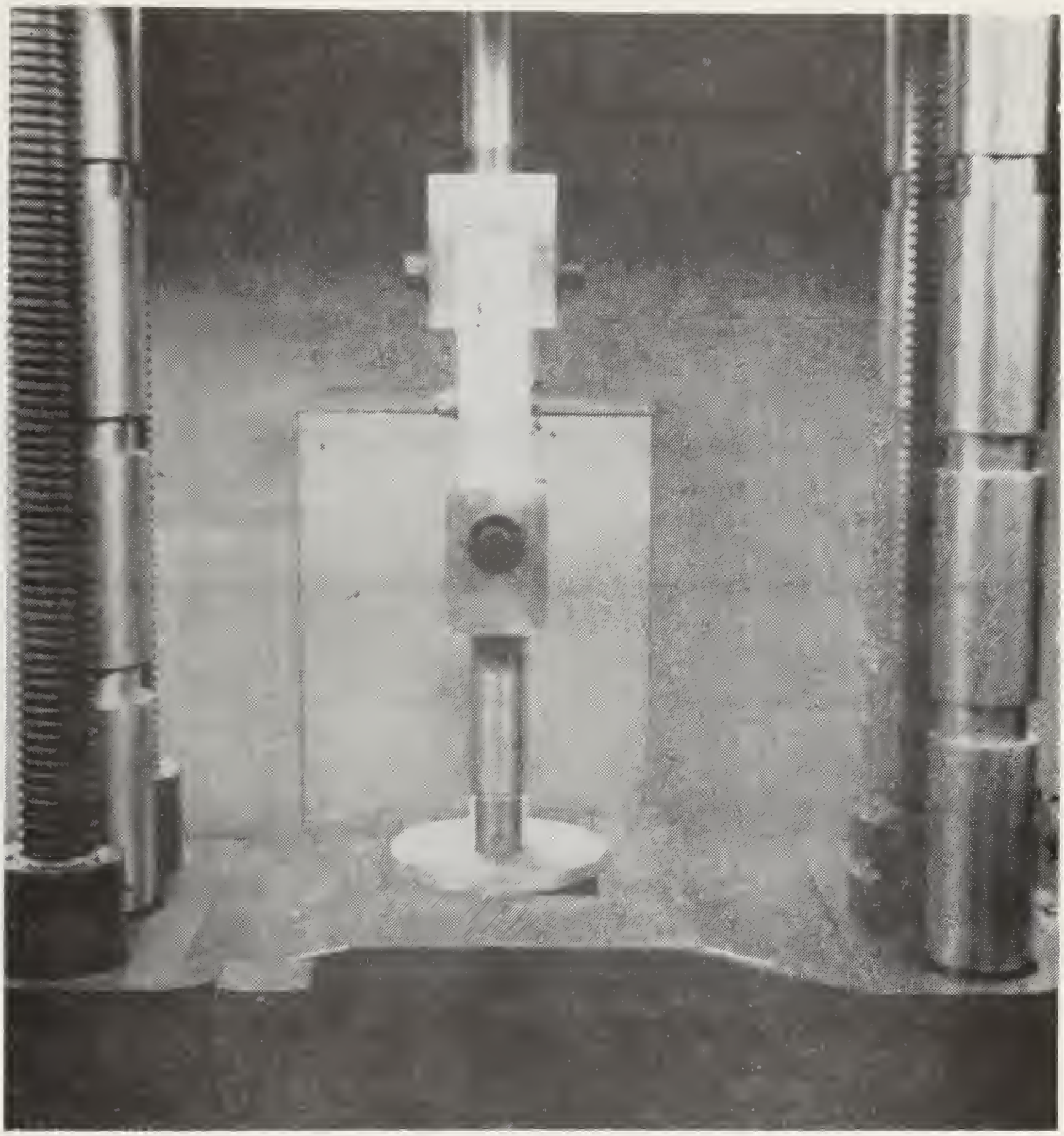

Figure 18. Riehle test machine crossheads with fixture attached.

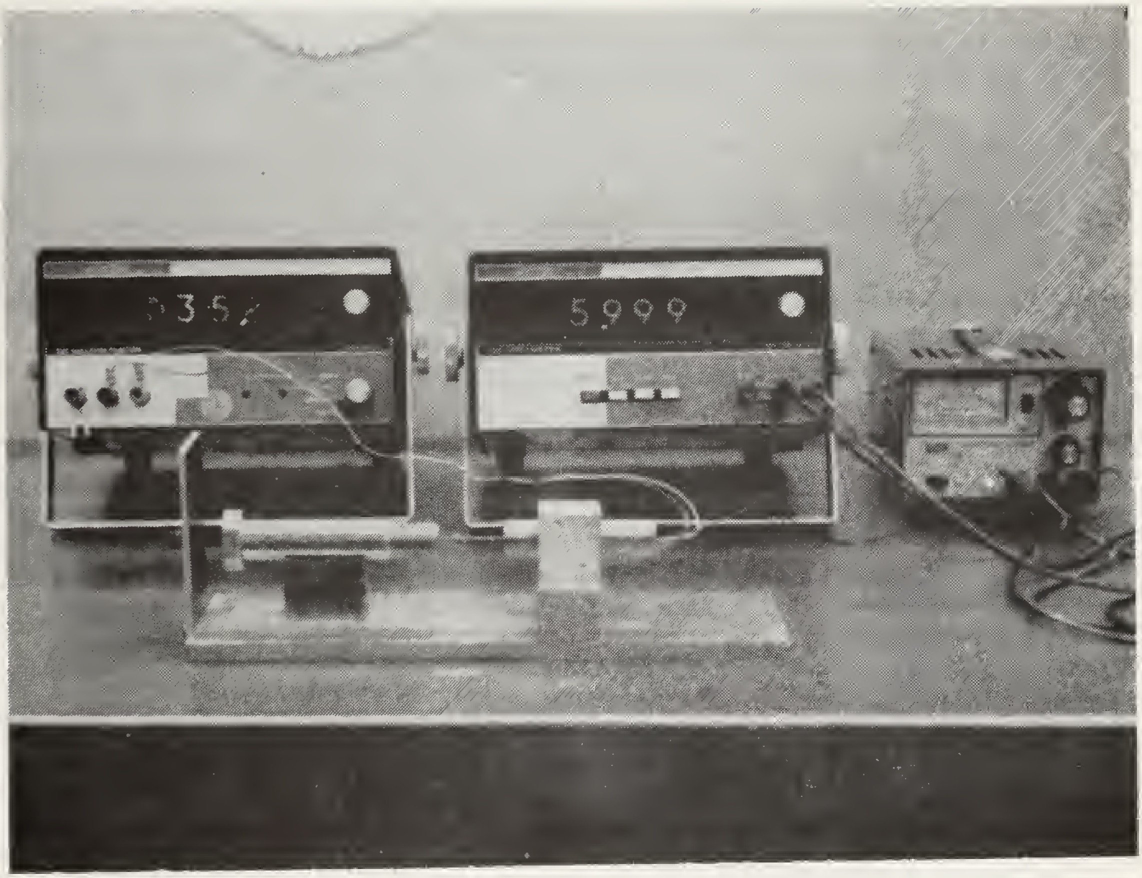

Figure 19. The measuring setup with the L.V.D.T. in the foreground.

4. <u>Test Fixture</u>

The fixture in Figure 20 was made out of 7075-T6 aluminum with steel insert where the bolt fits through. 7075-T6 was used because of its good machineability, relatively good strength, and easy access in a time of monetary constraint. The lower strength of the 7075 did require thicker sections than was desired in some cases but the possible transverse loading capability was still there. The fixture was made so that the primary loading on the bolt was shear to enable transverse movement of the bolt in the bolt hole.

5. <u>Test Bolt and Nut</u>

The bolt used in the tests is the NAS 1312-46, 3/4"-16, UNJF-3A threaded bolt actually used in the A-7 hook point - shank joint. The nut is a 42W1216, 3/4"-16, UNJF-3B threaded nut which is also used in the attachment joint.

C. RESULTS

Three series of tests were run on each bolt, with a different clearance between the bolt and the bolt hole wall. Each bolt undergoing testing was preloaded to the 350 lb-ft. torque required by Reference 3. The loads in all cases were applied as fast as the test machine would allow and one cycle was from zero load to the planned test load, then back to zero load again. Since it was decided that the number of force pulses the hook point would see would be between five to ten, it was thought that five to ten cycles would be the place to begin testing.

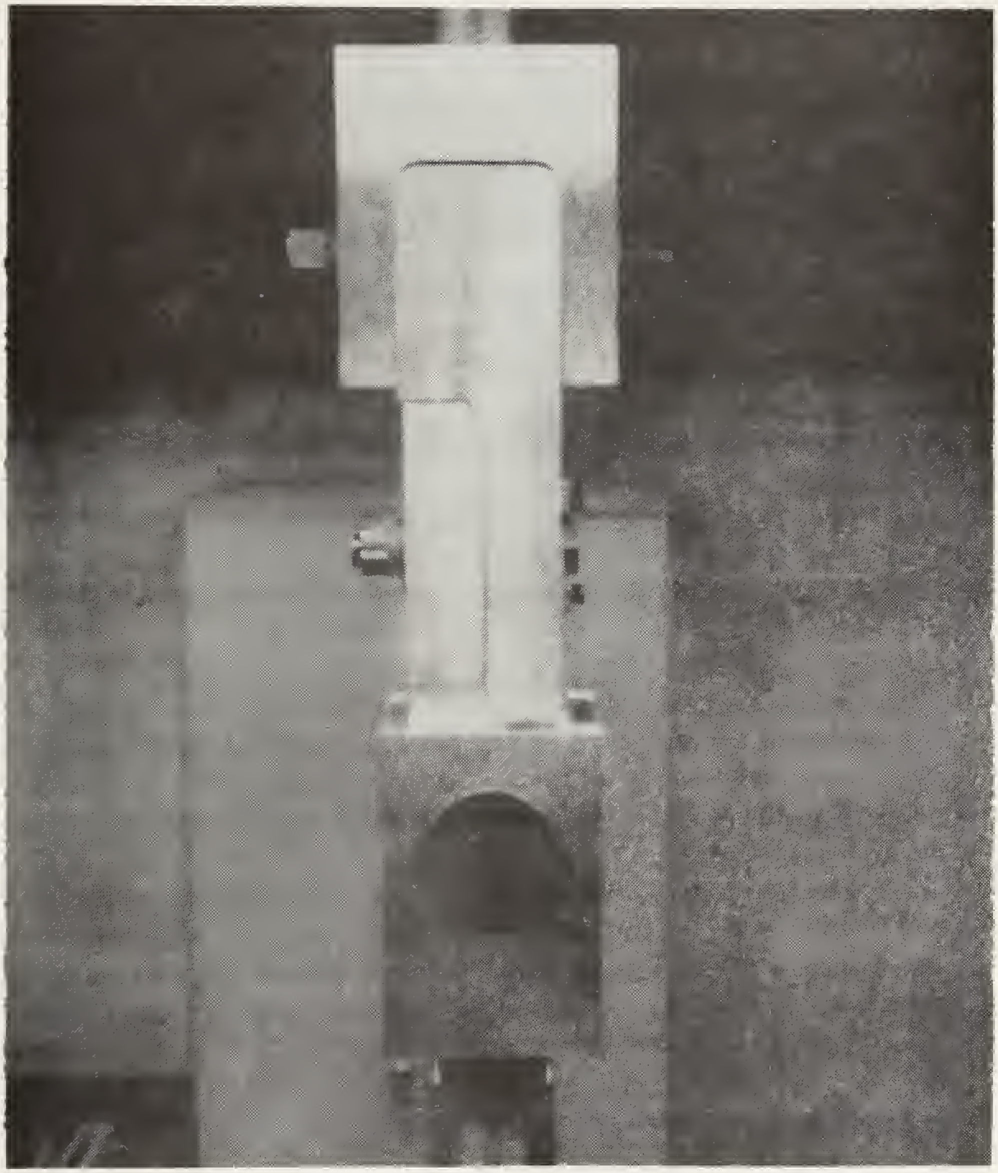

Figure 20. The test fixture.

In Series A, Table 3, the plan was to impose relatively high forces with only a few number of cycles, which was in keeping with the test design. The clearance used was the .003 inches that was actually measured from the shank and bolts that were on hand. The initial force used was 40-42,500 lbs. which was just slightly under the shear strength of the bolt, but since the load application was very slow the bolt deformed and length measurements were useless. Test A was run with different loads to a final 32.5-35,000 lbs. range, for five and ten cycles. In all cases the bolts had a torque loss, except bolt no. 9, however they were all plastically deformed as in Fig. 21, including the last case because one cycle went to 42,500 lbs. by mistake, and had to be cut or pressed out of the bolt hold and a new bushing installed. It was decided that the loads were too high and that for such a slow application of force, a larger tolerance between bolt and bolt hole diameter was necessary to have a better chance of the bolt "wobbling" in the bolt hole.

It was decided to continue testing in this manner because it was still considered possible to induce a very low frequency vibration or wobble in a larger bolt hole. Again according to Junker (Ref. 30), the frequency does not directly affect the self loosening mechanism, but the load's ability to cause relative motion. Thus if the hole were larger the bolt would tend to rock in the bolt hole while the nut would stay relatively parallel to the fixture face. In this manner it was hoped to cause relative motion and loosening without any extension.

TABLE III

SERIES "A" TEST DATA

Test No.	Bolt No.	Bolt* Test	Load Applied x 1000 lbs. Min.	Max.	Torque Loss	Number of Cycles	Comments
1	-	1	42.5	46.0	-	1	Failed in shear
2	-	1	40.0	42.5	50	10	Pressed out
3	-	1	40.0	42.5	40	5	Cut out
4	9	1	37.5	40.0	0	11	Galled threads requiring 100 ft-lbs of torque per turn to rotate nut off
5	-	1	35.0	37.5	50	10	Pressed out

*Seqential number of tests on each individual bolt.

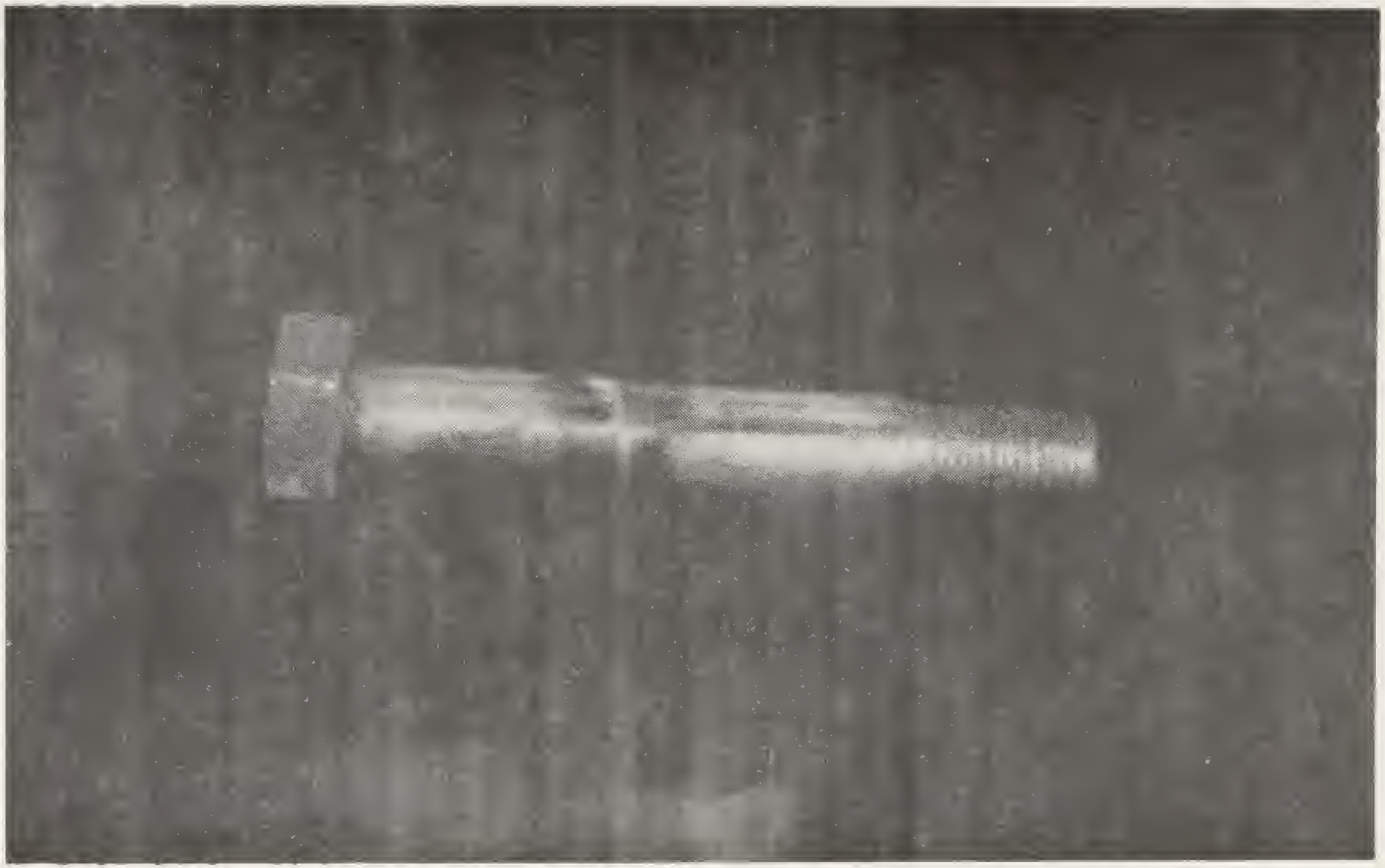

Figure 21. Bolt showing shear deformation incurred during testing.

Also if this did occur, it would be a significant finding because others have used varying force but the frequency of application and number of cycles were all much higher. Thus the test plan was changed by lowering the magnitude of the applied forces and increasing the number of cycles to a range of ten to twenty.

Another problem was discovered although not until four bolts were rendered useless for testing. This was the galling of the threads due to the leading nut thread overrunning the threaded portion of the bolt and crimping on the bolt shank. It was found that some bolts were longer than others and the shank portion would stick out beyond the fixture bolt hole edge. Thus the nut threads could run up onto the shank before the nut was fully torqued. This not only ruined the threads of each nut, but also created a locking mechanism which allowed little or no torque loss for a large amount of deformation. Examples in Test A are bolt number 9, with no loss, and in Test B, bolts no. 4 and 1 with no loss and bolt no. 2 with a relatively small loss for that amount of microstrain. The problem was not fully recognized until two more bolt-nut combinations were ruined in Test Series B. The problem was solved by putting a thin washer between the nut and the fixture.

In Series B, Table 4, the diameter tolerance was made .007 inches which is still far under the .0114 possible. The load levels were kept around the 30,000 lb. range and the cycles were varied from ten to twenty. There were still large

TABLE IV

SERIES "B" TEST DATA

Test No.	Bolt No.	Bolt* Test No.	Load Applied x 1000 lb Min.	Load Applied x 1000 lb Max.	Length of Bolt Inches Before	Length of Bolt Inches After	Microstrain x 10^6 in/in	Torque Loss ft-lbs	No. of Cycles	Comments Loads are x 1000 lb
1	2	1	35	37.5	4.2982	4.310	2745.3	50	10	Galled threads, 1 pull at 42.5 had to be cut out of fixture
2	5	1	30	32.5	4.2935	4.295	349.3	25	10	New washer
3	5	2	30	32.5	4.295	4.296	232.8	30	10	
4	6	1	30	33.25	4.292	4.2955	815.5	50	15	
5	3	1	30	32.5	4.2903	4.292	349.6	10	20	
6	1	1	32.5	35	4.2828	4.2852	560.4	40	10	
7	10	1	32.5	35	4.3040	4.3045	116.2	0	15	
8	4	1	32.5	35	4.284	-	-	0	10	Galled threads
9	1	2	27.5	30	4.2852	-	-	0	15	Galled threads
10	3	2	27.5	30	4.292	4.2975	1281.4	100	10	1 pull 42.5, New washer
11	6	2	30	32.5	4.2955	4.299	814.8	35	10	

*Sequential Number of Tests on Each Individual Bolt

amounts of microstrain associated with the torque losses, and the torque loss varied directly with the amount of microstrain. Therefore, it was judged that there was no self lossening present.

In Series C (Table 5) the clearance between bolt diameter and bolt hole diameter was increased to .010 inches. The loads were judged still to be too high, therefore Series C was started at 18,500 lbs. and graduated up to 32,000 lbs. The number of cycles were kept the same. There was immediately some excitement because some of the tests showed torque loss with very little or no measurable microstrain. Upon inspection however it was noticed that a new washer which had been made was indented where the bearing surface of the nut rested, Fig. 22,23. It was also noticed that the torque loss was less the next time the washer was used; bolts referenced are numbers 6, 3, and 11 in the first three tests in the series. To check the results another washer was made and used it on tests of the nest four bolts, numbers A, 10, G and 11. Approximately the same results, Fig. 24,25 occurred with the torque getting less each time the washer was reused except when there was significant microstrain present, as with bolt G. In series C there were some cases where torque loss occurred with little or no permanent microstrain present but not regularly enough to be considered self loosening.

TABLE V

SERIES "C" TEST DATA

Test No.	Bolt No.	Bolt* Test No.	Load Applied x 1000 lb Min.	Max.	Length of Bolt Inches Before	After	Microstrain $\times 10^6$ in/in	Torque Loss ft-lbs	No. of Cycles	Comments Loads are x 1000 lb
1	6	3	18.5	20	4.299	4.2992	46.5	20	10	New washer
2	3	3	20	22.5	4.2975	4.2976	23.2	10	10	
3	11	1	22.5	25	4.3045	4.3045	0	0	10	
4	A	1	22.5	25	4.2950	4.2951	23.3	30	15	New washer
5	10	2	22.5	25	4.3045	4.306	348.5	20	15	
6	G	1	22.5	25	4.306	4.3062	46.4	10	15	
7	11	2	25	27.5	4.3045	4.3055	232.3	10	10	1 Pull to 42.5
8	5	3	25	27.5	4.296	4.2963	69.8	0	15	1 Pull to 31.25
9	6	4	27.5	30	4.2992	4.300	186.1	10	10	1 Pull to 34.0
10	3	4	27.5	30	4.2976	4.2976	0	30	15	Bolt slightly bent
11	B	1	30	32.5	4.2885	4.2895	233.2	15	10	

*Sequential Number of Tests on Each Individual Bolt

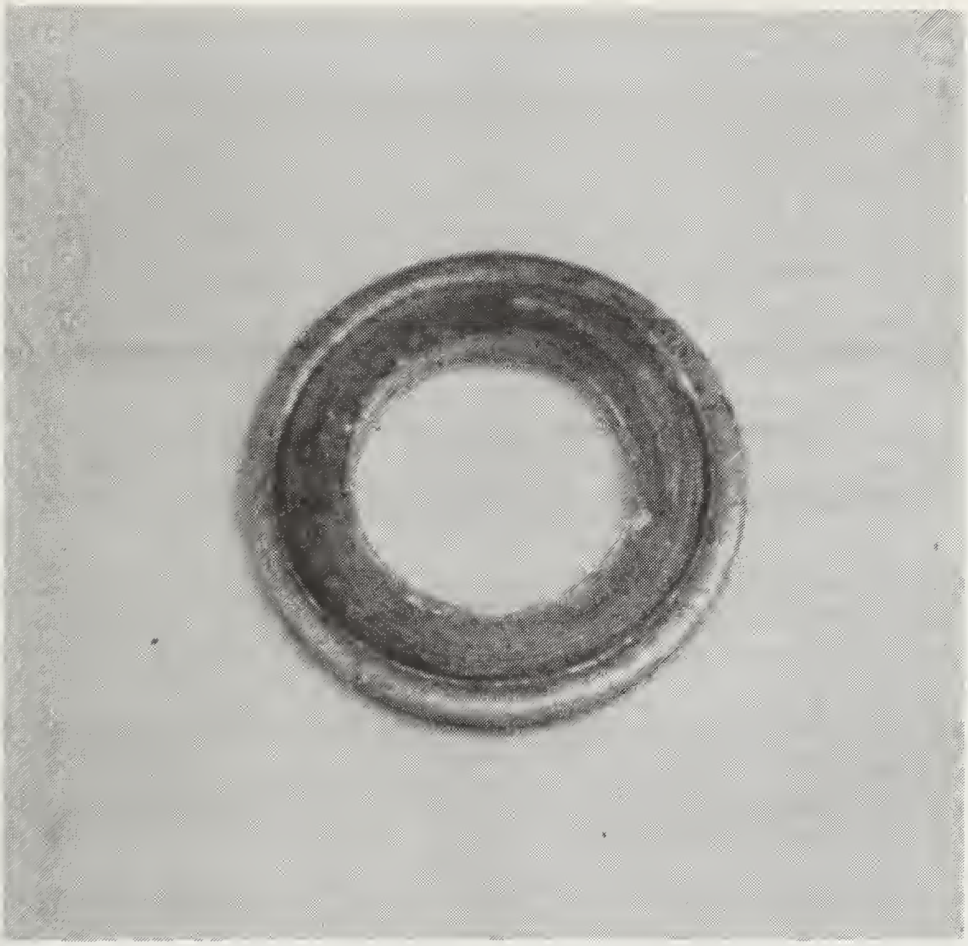

Figure 22. Crushed washer from first three tests in Series "C".

Figure 23. Nut seated in the crushed part of the washer.

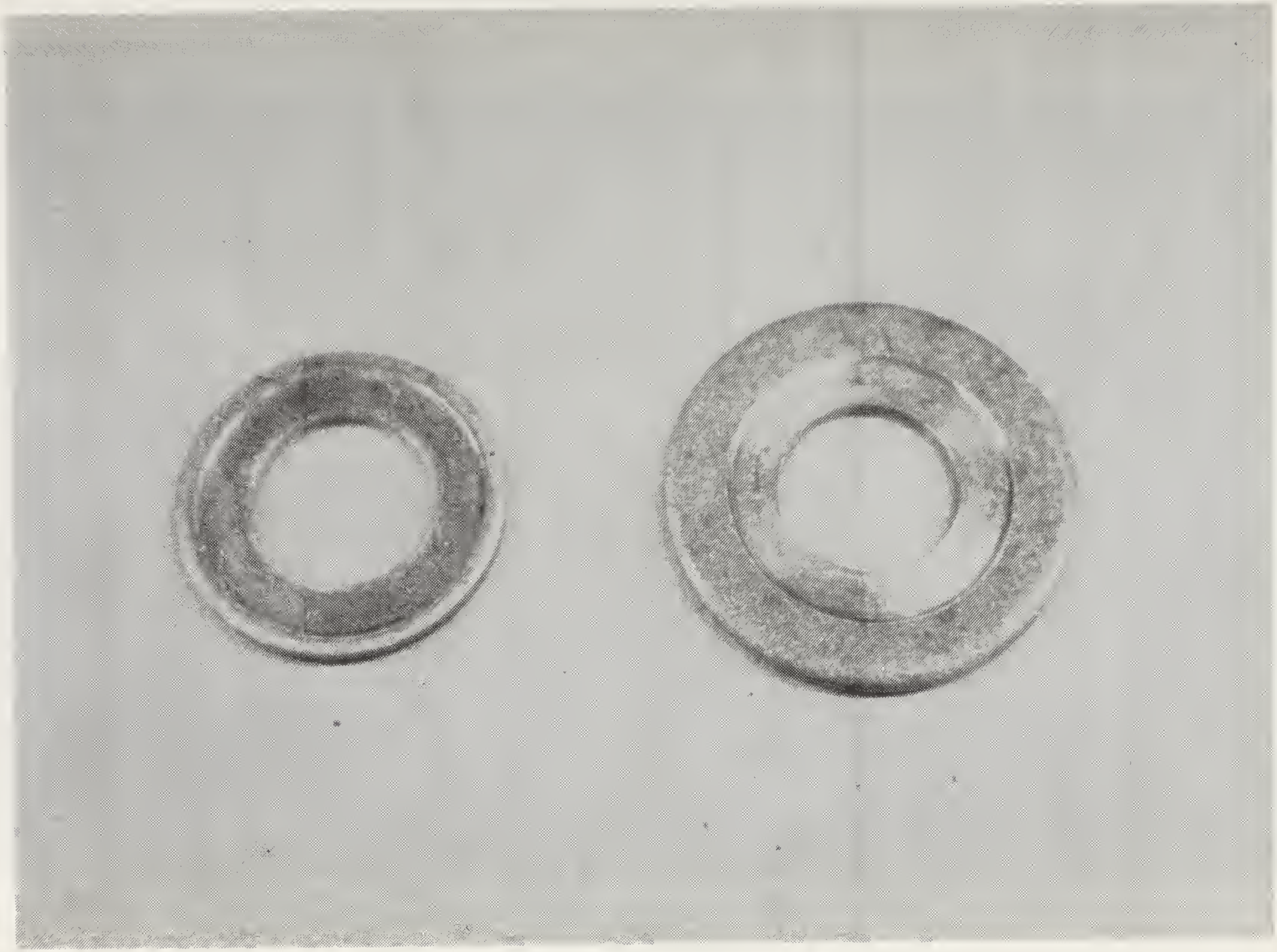

Figure 24. The two crushed washers in Series "C".

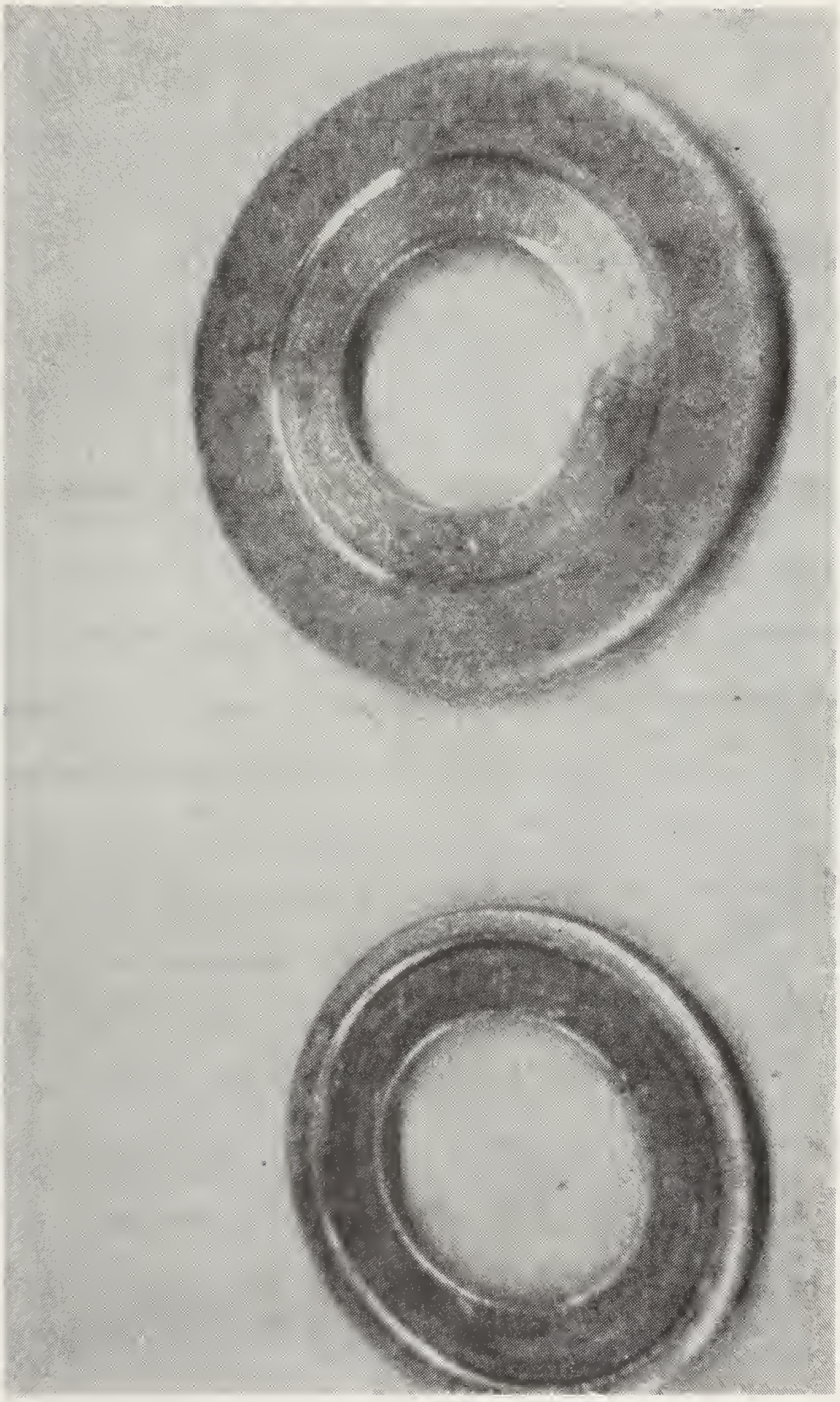

Figure 25. The two crushed washers in Series "C".

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

Even though the tests could not be performed as desired, the tests did show results that were relevant to the A-7 hook problem. First, it was shown that the bolt experiences permanent strain at low load levels. The tests made in this investigation were conducted at relatively low load levels compared with those involved in an actual arrestment evolution. NARF North Island reported (Ref. 7) that bolt failures could be initiated by a fore-and-aft rotation, which caused bending, tension, and shear load on the bolt. Since permanent strain accompanied every torque loss in the laboratory, and since it was produced at relatively low load levels, it is considered likely that the torque loss in fleet aircraft is due to deformation of the bolt rather than a self-loosening mechanism.

This conclusion is supported by the fact that the torque losses in the fleet do not occur with an consistency, but intermittently as in the VA-125 tests (Ref. 18,19). It also agrees with findings to be reported by the Naval Air Test Center, Patuxent River (Ref. 38), in which high speed motion photography of field arrested landings and carrier arrested landings showed that large torque losses occurred after the hook had struck rigid obstructions on the deck.

Another result pertinent to the A-7 problem is the performance of the washer, leading to loss of torque. The

Navy has been using a soft washer under the nut. Our tests, and tests to be reported by Patuxent River, indicate that the washer, if not hardened, will crush, causing loss of torque.

These tests have not proven that self loosening does not occur under the conditions present in the A-7 landing arrestment evolution: low frequency, low cycle, and large magnitude dynamic impact loadings. Junker (Ref. 29) cites an example of heavy impact axial loading tests conducted by E. G. Paland as producing nut rotation and loosening. Thus the possibility of such a mechanism contributing to the A-7 problem still exists: the tests reported here do not demonstrate the existence of such a mechanism, but neither can they be considered to preclude it.

It may be noted that the torque losses actually measured were not in exact agreement with those predicted using the method described in Section III - B. Figure 26 shows a comparison of the actual and predicted data. The reason for this discrepancy seems to be that in the development of the predicting equations it was assumed that the parts being clamped were rigid. This assumption ignores the actual elasticity of the fixtures, which would supply an additional spring constant and reduce the torque loss.

B. RECOMMENDATIONS

It is recommended that:

a. the high-strength bolts and nuts now being tested be supplied for fleet use.

b. only high-strength (hardened) steel washers be used in the hook retaining assembly.

c. further tests should be conducted to ascertain whether the self loosening mechanism described in this thesis is or is not a contributor to the loosening of the A-7 hook point retaining bolt. These tests should be conducted using a closed-loop electrohydraulic test machine, capable of accurately simulating actual arresting hook loads.

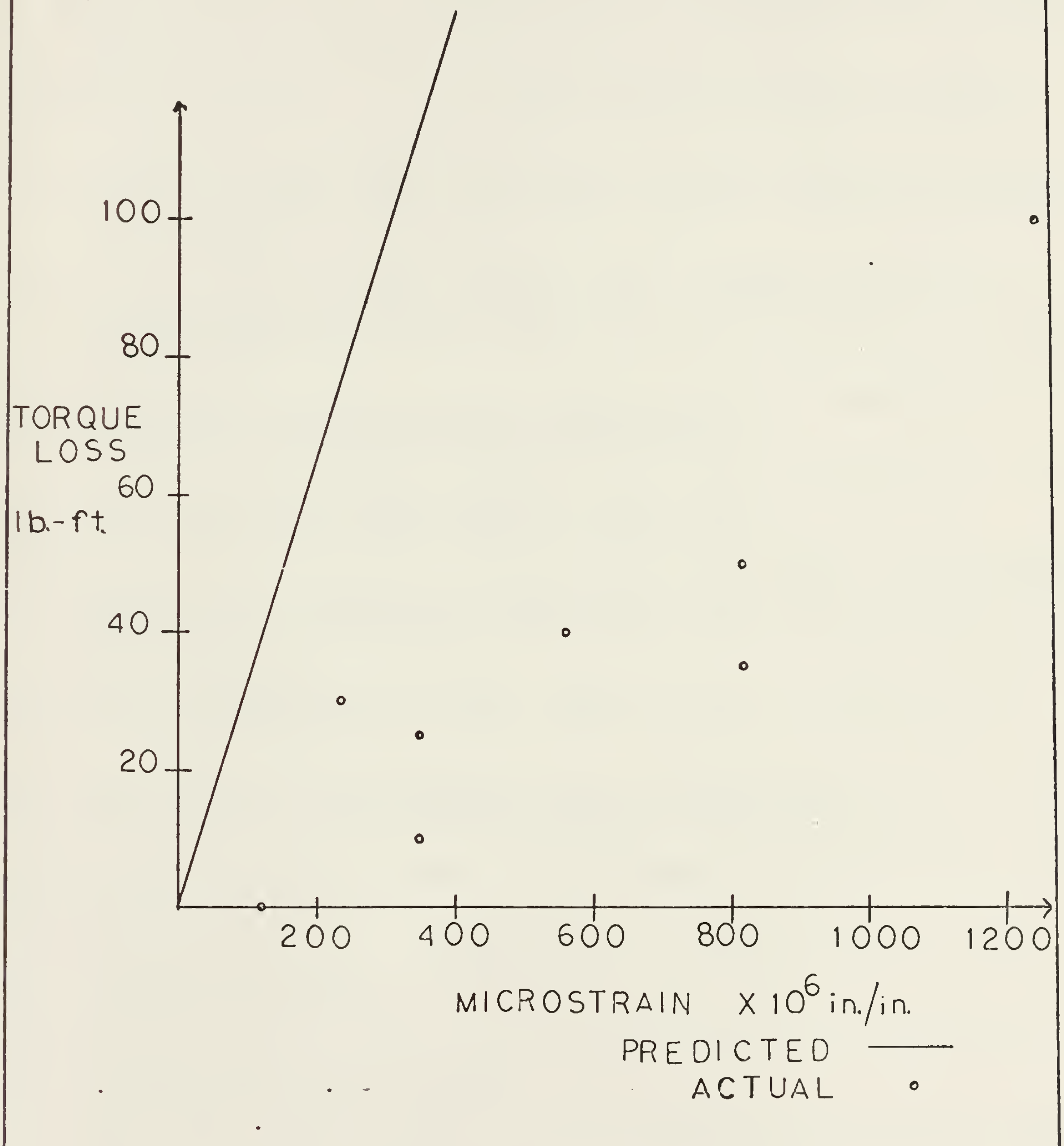

Figure 26. Comparison of predicted torques losses and actual torque losses versus microstrain.

REFERENCES

1. U. S. S. Kitty Hawk, Msg., 1408332 Mar 72, "Safety UR/ Request for Engineering Investigation".

2. COMNAVAIRPAC, Msg., 142122Z Mar 73, "Arresting Gear Bumper/ Hook Point Problems / Comments Concerning".

3. NAVAIR 01 45aae - 2 - 4, <u>Maintenance Information Manual</u>, A7 A/B/E.

4. NAVAIRSYSCOM HQ, Msg., 18 0338Z Apr 73, "Model A-7 Aircraft; Arresting Hook Attachment Bolt and Nut: Contract N00019-72-C-0078".

5. NAVAIRREWORKFAC NORTH ISLAND, Msg., 221844Z Nov 72, "Engineering Analysis A7E/157516 Hook Point Bolt Failure; Results of".

6. ATKRON ONE TWO FIVE, Msg., 160235Z Feb 73, "Safety UR/ Request for Engineering Investigation".

7. NAVAIRREWORKFAC NORTH ISLAND, Msg., 230144Z Feb 73, "A-7 Arresting Hook Point Attach Bolt".

8. Mentzell, J., private conversation, Code 360, A-7 Airframes Engineering, NARF JAX, Cognizant Field Activity for A-7 Aircraft, Nov 13, 1974.

9. NAVAIRSYSCOMREPLANT, Msg., 020332Z Jun 70, "Safety UR/ Incident Report".

10. NAVAIRENGCEN PHILADELPHIA, Msg., 112214Z Apr 72, "A-7 Hook Point Attachment Bolt Preliminary Report".

11. U. S. S. Coral Sea, Msg., 081032 May 72, "A7/F8 Hook Attaching Hardware".

12. COMNAVAIRPAC, Msg., 020156Z May 72, "Safety UR/Aircraft Incident Report".

13. NAVAIRSYSCOM HQ, Msg., 222144Z Nov 72, "A-7 Hook Point Matched Bolt/Nut Assy".

14. NAVAIRSYSCOMREPPAC, Msg., 042144Z May 72, "A-7/F-8 Arrest Hook Attach Hardware".

15. NAVAIRSYSCOMREPPAC, Msg., 142158Z Nov 72, "A-7 Arresting Gear Hook Point Attach Bolt".

16. COMNAVAIRPAC, Msg., 270246Z Jan 73, "A-7 Arresting Hook Point Attaching Bolt/Nut Assembly".

17. ATKRON ONE TWO FIVE, Msg., 160235Z Feb 73, "Safety UR/ Aircraft Incident Report/Request for Engineering Investigation".

18. ATKRON ONE TWO FIVE, Msg., 230150Z Feb 73, "A-7 Tail Hook Point Attach Bolt Summary".

19. ATKRON ONE TWO FIVE, Msg., 212240Z Feb 73, "A-7 Tail Hook Bumpers".

20. NAVAIRREWORKFAC NORTH ISLAND, Msg., 062254Z Mar 73, "A-7 Arresting Hook Point Attach Bolt/Nut".

21. NAVAIRSYSCOM HQ, Msg., 021646Z Mar 73, "A-7 Arresting Hook Attachment Bolt/Nut; New Design".

22. NAVAIRSYSCOM HQ, Msg., 182206Z Apr 73, "Arresting Hook Attaching Bolt/Nut; New Design A-7 Aircraft".

23. COMNAVAIRPAC, Msg., 200026Z Dec 73, "A-7 Carrier Suitability Problems SITREP No. 3".

24. COMNAVAIRPAC, Msg., 200023Z Dec 73, "A-7 Tail Hook Attach Bolt".

25. Goodier, J. N., "The Distribution of Load on the Threads of Screws," Journal of Applied Mechanics, Transactions A.S.M.E., Vol 62, p. A-11, 1940.

26. Hetenyi, M. "Photoelastic Study of Bolt and Nut Fastening," Journal of Applied Mechanics, Transactions A.S.M.E., June 1943.

27. Goodier, J. N. and Sweeney, R. J., "Loosening by Vibrations of Threaded Fasteners," Mechanical Engineering, Vol. 67, No. 12, Dec 1945.

28. Sweeney, R. J., "How to Avoid Loosening of Bolts and Nuts," Machine Design, Vol 18, No. 6, June 1946.

29. Junker, G. H., "Criteria for Self Loosening of Fasteners under Vibrations," Part I, Aircraft Engineering, Vol 44, No. 10, pp. 14-17, October 1972.

30. Sauer, J. A., Lemmon, D. C., and Lynn, E. K., "Bolts How to Prevent Their Loosening," Machine Design, Vol 22, No. 8, August 1950.

31. Clark, S. K., and Cook, J. J., "Vibratory Loosening of Bolts," Society of Automotive Engineers Paper No. 660432, Meeting of June 6-10, 1966.

32. Junker, G. H., "Criteria for Self-Loosening of Fasteners under Vibrations," Part II, Aircraft Engineering, Vol 45, No. 1, pp. 12-19, Jan 1973.

33. Junker, G. H., "Criteria for Self Loosening of Fasteners under Vibrations," Part III, Aircraft Engineering, Vol 45, No. 2, Feb 1973.

34. Guy, A. G. and J. J. Hren, Elements of Physical Metallurgy, Addison-Wesley 1974.

35. Timoshenko, S.and D. H. Young, Elements of Strength of Materials, Fifth Edition, D. Van Norstrand Company, Inc., 1968.

36. Naval Engineering Center, Performance Analysis of the Mark 7 Mod 3 Recovery System Based on Aircraft Calibration Tests, Aug. 1, 1969.

37. Vought Aeronautics Co., (L.T.V. Aerospace), Engineering Drawings, Contract No. 66-0003.

38. Wilson, T. E., Private Communication, Code FT2222 Naval Air Test Center, NAS, Paxtuxent River, Md.

DISTRIBUTION LIST

		No. of Copies
1.	Defense Documentation Center Cameron Station Alexandria, Virginia 22314	2
2.	Library, Code 0212 Naval Postgraduate School Monterey, California 93940	2
3.	Department Chairman Department of Aeronautics Naval Postgraduate School Monterey, California 93940	1
4.	Professor M. H. Bank Department of Aeronautics Naval Postgraduate School Monterey, California 93940	1
5.	Commander N. B. Endo Code AIR-5102L COMNAVAIRSYSCOM Naval Air Systems Command Headquarters Washington, D. C. 20360	1
6.	Mr. V. D. Elko Code AIR-436 COMNAVAIRSYSCOM Naval Air Systems Command Headquarters Washington, D. C. 20360	1
7.	Commander J. C. Clinton Code 724 Commander Naval Air Force; Pacific Fleet NAS North Island San Diego, California 92135	1
8.	Mr. T. E. Wilson Code FT 2222 Naval Air Test Center NAS Patuxent River Lexington Park, Maryland 20670	1
9.	LT A. W. Roessig Jr. 5832 Walnut Omaha, Nebraska 68106	3
10.	LCDR R. F. Johnson VF-24 FPO San Francisco, California 96601	1

Thesis 157042
R6785 Roessig
c.1 Investigation of the
A-7 hook point attachment bolt loosening
problem.